9

AD679449

B·S·R·A

JOURNAL *of Abstracts*

OF THE

British Ship Research Association

Volume 23 Number 9

SEPTEMBER 1968

ABSTRACTS NO.

26 652 — 26 702

THE BRITISH SHIP RESEARCH ASSOCIATION

Prince Consort House,
27-29 Albert Embankment,
London, S.E.1.
Telephone: RELiance 6481.
Telex: 261952.
Telegraphic Address:
Templace London Telex.

Wallsend Research Station,
Wallsend,
Northumberland.
Telephone: Wallsend 62-5242.
Telex: 53476.
Telegraphic Address:
Templace Wallsend Telex.

44

THE RESEARCH COUNCIL

ABSTRACTS No. 26,652 — 26,702

JOURNAL *of Abstracts*

OF THE

British Ship Research Association

Editor: R. C. KAHLER

Volume 23 No. 9 September 1968

TABLE OF CONTENTS

CLOSURE OF B.S.R.A. LONDON OFFICE

The London Offices of B.S.R.A. are to be closed on Friday, 8th November 1968 and from Monday, 11th November, all activities will be conducted from the Wallsend Research Station.

FOR USE OF MEMBER FIRMS

Member Firms can usually borrow the originals of any of the Abstracts, or photographic copies can be provided. In addition the Association will supply translations of foreign articles. Requests for these services, which are free of charge, should be made on the form below. Only the serial numbers (printed in bold figures) of the Abstracts need be quoted.

To	*From*
Head of Technical	....
Information Division	
B.S.R.A.,	....
Wallsend Research Station,	
Wallsend,	....
Northumberland.	
	....
	Date ....

Mark reply for the attention of ....

Please send me, ON LOAN, the originals of the following Abstracts:
Serial numbers:—

....	....	....
....	....	....
....	....	....

Please have the originals of the following Abstracts TRANSLATED for me:
Serial numbers:—

....	....	....
....	....	....
....	....	....

Please make and supply a COPY OF THE ORIGINALS of the following Abstracts, which I require for the purposes of research or private study. I have not previously been supplied with a copy of these originals by any librarian. I undertake that, if a copy is supplied to me in compliance with this request, I will not use it except for the purposes of research or private study.

Signature...

Serial numbers:—

....	....	....
....	....	....
....	....	....

ENQUIRY SERVICE

The large amount of published information that has been accumulated in the Technical Information Division of B.S.R.A., and the contacts we have established with other organisations, make it possible to provide answers to many technical problems. Member Firms are invited to make use of this service, which is confidential. Enquiries may be made by telephone or letter; if they are written in the blank space below, please insert your name and address on the reverse side of the form.

ABSTRACTS *from Current Technical Literature*

The following Abstracts purport to be fair summaries of the articles, but the Association does not accept responsibility for statements made in the originals, nor does it necessarily agree with their contents.

The standard form of reference to the source of each Abstract is: Title of Periodical or Publication (abbreviated according to the list on pp. 3–19 of B.S.R.A. Journal for January 1968), volume number (in heavy type), year, and page number, followed by the date of issue where appropriate. The length of the article and other bibliographical details are also indicated.

SHIP RESISTANCE AND FLUID MOTION

(*See also Abstract No.* 26,667)

26,652 **The Hydrodynamic Resistance of External Hull Anodes.** GOULD, R. W. F., and BOWDEN, B. S. *Shipp. World & Shipb.*, **160** (1967), p. 1721 (Oct.) [4 pp., 2 ref., 5 tab., 4 graphs, 3 diag.]

The work described in this article was carried out in the 7 × 7 ft wind tunnel of the Aerodynamics Division of the N.P.L. on behalf of B.S.R.A. It constitutes a sequel to an earlier investigation, in which B.S.R.A assisted, into the effect on performance of fitting anti-corrosion anodes to the hull of the 18,000-ton d.w. tanker *San Fortunato* (see Abstract No. 25,045, Feb. 1967). At the time of the earlier investigation, little information was available on the hydrodynamic resistance of anodes.

Two models of the design of anode (" No. 1 ") fitted to the *San Fortunato* were tested; one had a smooth surface, whereas the other was coated with Derlex grit to represent an anode surface after a period of service. A third model was also tested; this was of different design (" No. 2 ") and had a smooth surface. The resistance was measured for each anode when positioned at angles to the stream flow ranging from zero to 30° in increments of 5°. The results are presented in the form of resistance coefficients and cross-flow force coefficients, and they are plotted against the stream-flow angle.

When calculating the increase in resistance due to fitting anodes to a hull, it is recommended that the velocity used in association with the resistance coefficients obtained from the tests should be the local velocity in the ship's boundary layer at the outer surface of the anode. The procedure is illustrated in an appendix; two methods are used to calculate the effect on ship performance of the anodes fitted to the hull of the *San Fortunato*. For a ship speed of 14·5 knots, the difference in measurements during the two series of trials (see again Abstract No. 25,045) was consistent with an increase in power requirement due to the anodes of the order of 50 s.h.p. The calculated increase is in good agreement with this, assuming that the anodes were aligned to within 5° of the stream flow.

The results of the tests show that the effect of not aligning anodes with the stream flow will be reduced if they are designed with rounded end-profiles. " No. 2 " anodes will therefore have lower resistance coefficients than " No. 1 " anodes, and should not be subject to separation of the stream flow provided that they are aligned to within 15° of the stream-flow direction.

26,653 **Model Testing in Ice.** VANCE, G. P. *Naval Engrs J.*, **80** (1968), p. 259 (Apr.) [6 pp., 9 ref., 8 graphs, 7 phot.]

After a study of earlier attempts at model testing of icebreakers, and extensive theoretical work on the scaling laws to be observed (see also Abstract No. 26,205, Mar. 1968), the Icebreaker Design Branch of the U.S. Coast Guard concluded that satisfactory model testing should be possible using ice sheets grown under closely controlled conditions. The only suitable tank in which such close control can be maintained is the Arctic Pool of the Naval Underseas Warfare Center, San Diego, California; this was designed and equipped for investigations on ice, not on ship models. However, the Coast Guard set up model-test equipment and ran a "Wind" class icebreaker model in March 1967. This feasibility test confirmed that consistent results were obtainable in fresh-water ice sheets thicker than about 0·2 in (thickness could be controlled to within 0·05 in); the failures were tensile, as on full scale. Model results for continuous breaking were more optimistic than full-scale data and theoretically-based predictions; this was expected from scaling considerations which are explained.

In July 1967 three new triple-screw hull forms were model-tested in the Pool. One (M-5) had a concave stem line, fairly flat sides, and a special stern with broad sections above the propellers (the intention being to exclude ice). Another (M-9) was a modification of the *Glacier's* hull form (see Abstract No. 15,506, July 1959), and the third (M-7) was a compromise between the first two, retaining a concave bow and some ice-exclusion properties but having an icebreaking stern above and slightly below the waterline. Resistance and propulsion testing of the three forms was conducted at the Naval Ship Research and Development Center, Washington. The models are shown in photographs; the results of "ramming" and "continuous" icebreaking tests (converted to full scale) are presented graphically for M-5 and M-9 and compared with theoretically predicted curves. These graphs relate ice thickness broken to impact velocity (ramming) or to s.h.p. and ship speed (continuous breaking); the horizontal and vertical accelerations at impact are also shown on a base of impact velocity. Further graphs (including results for M-7) show the horizontal accelerations on impact when backing at different speeds, and the power/speed relations when running in ice-clogged channels.

Ice flow around the forms was studied by ciné camera and underwater television. As a result of the tests, a modified M-5 form is being developed for a new Coast Guard icebreaker. It is noted that work on icebreaker model-testing is being hampered by a lack of systematic full-scale data for correlation purposes.

PROPELLERS AND PROPULSION

(*See also Abstract No.* 26,667)

26,654 **Considerations on Propeller Layout from the Engine Builder's Point of View.** SMIT, J. A. *Instn E. Shipb. Scot., Paper No. 1329, presented* 27 Feb. 1968 [13 pp., 3 ref., 5 graphs, 1 diag.]

The engine builder assesses the performance of a propeller from engine-room log-book records of r.p.m., load, fuel consumption, etc. These are

more reliable in ordinary service conditions than estimates of ship speed through the water on which the naval architect bases his assessment of the propeller. Working conditions incompatible with the propeller design lead to increased wear and maintenance costs of the machinery, and even to possible eventual breakdown. Service results over the last 5 to 10 years have shown that loss of r.p.m. over a period of operation may be considerable, especially in large vessels with high block coefficients.

The service conditions for a fixed-pitch propeller are demonstrated by a graph of engine power as a percentage of the design rating (N) against r.p.m. as a percentage of the design value (n). If the propeller is designed so that the combination of N and n corresponds to trial conditions of the fully-loaded ship with a clean new hull in smooth water, it will be found that this propeller will be too "heavy" for actual service. Increase of hull resistance by fouling and surface deterioration will cause a drop in r.p.m., with consequent loss of power, along a line on the graph representing constant mean effective pressure. This loss of power occurs in conditions for which more power is needed to drive the ship at a given speed. Although the engine will be producing less power and consuming less fuel, its specific thermodynamic and mechanical loading will be increased.

The Author suggests that the propeller should be designed to absorb not more than 85-90% of the maximum continuous engine rating for the fully-loaded clean ship in trial conditions in smooth water. With a propeller so designed it may be impossible to fulfil some of the trial conditions stipulated in present-day building contracts. However, the Author considers that it should be acceptable for the trials to show that, at service speed, sufficient margin in output, torque, and r.p.m. has been provided by the engine builder. *In most instances, full-load output with* the clean ship in ballast condition will be obtained by increasing the r.p.m. to about 106% of the nominal speed.

Graphs are given which show, on a base of r.p.m., the progressive increase in required power (due to hull fouling, etc.) for a series of 70,000-ton d.w. tankers and for a series of 12,200-ton d.w. cargo liners. For the tankers, the excess engine loads after 4½ months, 8 months, and 13 months in service were 2, 10, and 22% respectively at nominal engine speed. The excess engine loads for the cargo liners were 4, 9, 15, and 17% after 3½, 5, 6½, and 8½ months. Even after dry-docking, the initial performance of the tankers was not restored. Another graph shows that the percentage increase of thrust required to maintain constant ship speed for an 11,000-ton d.w. cargo ship amounted to about 17% after five years. The percentage increase in resistance seems also to be a function of the size of the ship; analysis of service records shows that, after five years' service with regular dry dockings, the increase in engine output required to maintain constant ship speed ranges from about 20% for a 10,000-ton d.w. tanker or bulk carrier to about 42% for one of 70,000 tons d.w.

Van Lammeren *et al* recommend that, for a motor ship, the design r.p.m. of the propeller should be ½ to 1% above the engine r.p.m. at m.c.r. as quoted by the engine builder, to allow for the mean service condition. The Author considers that, for large bulk carriers and tankers, this margin should preferably be 4 or even 5%.

The increase in required power due to deterioration of the hull surface may be partly overcome by "correcting" the propeller. The simpler

of the two methods available is to crop the blades; this is effective only where there is no pitch reduction over the outer radii. A working rule is that 1·5% diameter correction gives 1 to 1·2% r.p.m. increase with constant torque. The limit of blade cropping is about 10% of the diameter. The other method, correction of pitch, is also limited to about 10% and can only be done in a specialised workshop. With combined cropping and change of pitch, the maximum increase in r.p.m. that can be expected with constant torque is 13–15%.

Finally, the Author briefly summarises the advantages of controllable-pitch propellers, and gives a sketch of a schematic arrangement for combined control of engine speed and propeller pitch; this allows the engine to operate at all times under the most favourable conditions.

26,655 **The Cavitation Laboratory at Skipsmodelltanken—General Description.** ANDERSEN, J. A., FRANCK-PETERSEN, R., and others. *Proceedings of Symposium on Testing Techniques in Ship Cavitation Research, Trondheim*, 31 May–2 June 1967; in *Norwegian Univ. Technol., Ship Model Tank, Publication No.* 99 (Dec. 1967), Vol. I, p. 1 [66 pp., 18 ref., 1 tab., 10 graphs, 21 diag., 16 phot.; and Discussion: 4 pp.] See also *Shipp. World & Shipb.*, **160** (1967), p. 1547 (Sept.), and p. 1744 (Oct.) [8 pp., 1 ref., 1 tab., 10 graphs, 10 diag.]

This paper explains how the model-propeller testing techniques to be used determined the design of the new cavitation tunnel (No. II) of the Norwegian Ship Model Experiment Tank, Trondheim. It is part of the symposium held in 1967 at the inauguration of the new tunnel and laboratory. (See also Abstract No. 26,309, Apr. 1968.) The tunnel plant itself, and the equipment and instrumentation, are described, and there are some nomograms relating various model-propeller parameters.

The *Shipp. World & Shipb.* article is an abridged version of the paper.

SHIP PERFORMANCE, STABILITY, AND MANOEUVRABILITY

(*See also Abstract No.* 26,667)

26,656 **Motions of a Ship in a Head or Following Wave.** GRIM, O. K. *Hamburg Model Basin (the HSVA), Report No. 1303* [30 pp., 8 ref., 8 graphs, 3 diag.]

This Report was prepared for the U.S. Department of the Navy.

Although a number of solutions, giving results which are in satisfactory agreement, are known for the hydrodynamic problems connected with the harmonic heaving motion of a two-dimensional body, there appears to be no equivalent solution for the hydrodynamic problems relating to the motions of a three-dimensional body in calm water or in a head or following wave. In some cases, the strip method or that of Korvin-Kroukovsky (see Abstract No. 13,519, Dec. 1957) may be satisfactory, but it is difficult to establish the limits for their application. A more precise method was described by the Author in 1960 (see Abstract No. 17,370, May 1961), but this was developed for zero speed only; for a finite forward speed many difficulties arise, and in particular the velocity potential for the harmonic flow becomes very complicated.

In the present Report, the Author presents a method of calculation, by

computer, for a body with zero speed or with a finite speed. The computer program, which is used with the Telefunken TR4 computer at Hamburg University, is suitable for any hull defined by ten or more sections. Its uses include the calculation of forced heaving or pitching motion in calm water.

The method provides results for:—

(*a*) the motions of the body in an incoming head or following wave,
(*b*) the harmonic motion of the free water surface,
(*c*) the harmonic relative motion between the body and the free water surface at the surface of the body,
(*d*) the shear force and bending moment,
(*e*) the distribution of the hydrodynamic force over the length of the body,
(*f*) the deformation of an incoming wave by a non-heaving and non-pitching body.

Some results for the simple ship-like form mentioned in Abstract No. 17,370 are shown in curves, and are discussed.

Results obtained with the aid of this computer program show reasonable agreement with experimental values, and are considered to be sufficiently accurate for any hull in any regular head or following sea.

26,657 **Horizontal Fins on Rudders Improve Directional Stability.** *Maritime Reporter*, **29** (1967), p. 32 (1 Nov.) [1 p., 1 diag., 4 phot.]

High block coefficients and low L/B ratios are characteristics of modern large tankers. This has not adversely influenced turning ability at sea, but it has had a detrimental effect on directional stability. Methods of improving directional stability, even at the expense of turning ability, have been studied by Kawasaki Dockyard. Existing special-rudder designs were not considered suitable for this application, and a series of model tests on rudders of different configurations was therefore conducted at Hiroshima University. First results showed that directional stability is improved if rectangular-shaped fins are fitted in a horizontal plane to the upper and/or lower edges of the rudder; the effect is the same as that of increasing the rudder height/breadth ratio.

A study of flow past the rudder blade, both under open-water conditions and with a propeller (but no hull), showed that the fins reduced the circulation loss. The arrangement of the test-rig equipment is shown in a system diagram. Rudder force and torque were measured by strain gauges, and fluid speed by a pitot tube. Ten different shapes of fin were prepared for the tests; there are photographs of the model rudder with and without fins.

The principal results are stated and discussed. Particular attention was paid to the influence of the propeller race (for both a right-hand and a left-hand turning propeller) on fin effectiveness over a range of rudder angles to port and starboard. The asymmetry due to the race can be considerable, especially if the rudder area above shaft level differs greatly from that below it. Large fins, whether attached above or below, increase normal force at all angles up to that of stall, which is not influenced. Accurate torque measurements were not possible due to vibration in the test rig, but trends were measured. The addition of fins changed the torque characteristics and also the centre of pressure; for some rudder

angles the torque was greater with fins than without, but at no time did it reach the maximum values obtained without fins. No increase in steering-gear capacity should therefore be required.

It is considered that more model and full-scale tests are required before optimum design methods can be evolved. In the meantime a finned rudder (shown in a photograph) has been fitted to the 100,000-ton d.w. tanker *Golar Nor* (see Abstract No. 23,854, Jan. 1966), and trial results are awaited. Patent rights have been applied for.

26,658 **Transverse Strength of a Large Steel Frigate Model.** CLARKSON, J., and WALLACE, G. *R.I.N.A., paper read* 23 Mar. 1967 [30 pp., 12 ref., 10 tab., 34 graphs, 14 diag., 8 phot.]

A detailed account is given of extensive experiments on a 69 ft long by 24¾ ft wide frigate-hull portion, constructed (with realistic details) in welded mild steel. This model (almost full scale) was tested in the N.C.R.E. Large Testing Frame under transverse loading without significant longitudinal stresses. The main objective was to obtain experimental confirmation of a three-dimensional elastic transverse-strength analysis by Yuille and Wilson (see Abstract No. 16,454, May 1960), and to establish values for empirical parameters in the idealisation of the structure. Good agreement was obtained between the measured and calculated deflections and stresses for both concentrated and distributed (hydrostatic) loads within the elastic range. Yuille and Wilson's curved-grillage method of analysis proved more accurate than conventional ring-frame analysis. The model was finally loaded to collapse under a uniform pressure applied to the bottom. The failure was influenced by a number of shortcomings in the detailed design, such as insufficiently stocky webs of the transverse frames, and intermittent welding of webs to flanges in fabricated stiffeners. The observations made illustrate the decisive influence of local and detail design on the collapse load; there is probably a greater need for improvements in this field than for more refined methods of elastic analysis.

Numerous diagrams of stress and deflection distributions (with measured points and calculated curves) are reproduced. There are photographs of the loading arrangements and of failures at structural details.

26,659 **The Prediction of Long-Term Distributions of Wave-Induced Bending Moment from Model Tests.** COMPTON, R. H. *Marine Technology*, **5** (1968), p. 137 (Apr.) [12 pp., 14 ref., 1 tab., 13 graphs]

The Author presents a straightforward procedure for predicting long-term distributions of wave bending moment, using the results of regular-wave model tests in conjunction with a suitable family of sea spectra. Specimen calculations for a typical tanker in a well-defined but non-typical wave environment show that reasonably accurate predictions can be made using a relatively small (9%) random sample from a large population of sea spectra, if the sea spectra are stratified by wave height and the distribution of wave heights is known. The procedure can, in principle, be used for design-stage predictions, though its validity will have to be confirmed by correlating predictions with full-scale data. Realistic wave distributions for the ocean areas of interest would also be necessary.

See also Abstract No. 26,315 (Apr. 1968).

SHIPBUILDING (GENERAL)

26,660 **Lloyd's Register's Rules for Dry-Cargo Ships.** ROBERTS, W. J. *Instn E. Shipb. Scot., Paper No. 1327, presented* 15 Dec. 1967 [60 pp., 15 ref., 2 tab., 14 diag., 4 graphs]

The Author discusses some of the reasoning on which the recent Lloyd's Register Dry Cargo Ship Rules (as given in Notice No. 2, 1967) have been based. These Rules differ from earlier ones in that scantlings are determined from formulae and not from tables. Nevertheless, their bases are not purely theoretical; some of the formulae have been modified empirically to take account of service experience with both successful and unsuccessful ships. In general, the application of the new Rules leads to greater structural efficiency rather than to radical reductions in steel weight. An important feature is the framing of the Rules in such a way that they lend themselves to easy programming for computer use.

These new Rules apply to ships of 90 m (295 ft) length and over. They are chiefly concerned with two aspects of ship structures, namely, longitudinal-strength and local-strength standards. The Author deals mainly with the second aspect. He considers only briefly the application of the longitudinal-strength standards, which have been altered to conform with those now in force for tankers; the principles involved have been discussed in earlier papers (see Abstracts No. 23,307, Aug. 1965, and 23,527, Oct. 1965). He proceeds to discuss in turn the following Sections of the new Rules.

Section 4—Deck Plating. The formulae for minimum deck thicknesses generally represent existing practice, which has proved satisfactory. Details have been included of elliptical openings on strength decks and of the edge reinforcement required around circular openings. Minimum plate thicknesses are given for use with fork-lift trucks of all-up weight up to 14 tons and having either two or four wheels at the fork end. The quality of material required on the decks of refrigerated ships has been slightly revised for working temperatures below $-20°$ C ($-4°$ F).

Section 5—Shell Plating. Shell plating forms a large part of the cross-section of the hull girder, which must have a minimum section modulus as laid down in Section 3 (Longitudinal Strength). This modulus requirement is thus a major factor in defining the thickness of the shell plating. Because of interaction between the shell and other parts of the ship's structure, the shell thickness cannot be satisfactorily predicted by simple theory. The formulae given in the new Rules include one that takes account of the relatively high shear forces that can occur in the side shell of certain ships, such as bulk carriers loaded in alternate holds; the permissible still-water shear stress varies with longitudinal position.

Section 6—Longitudinal Framing. Semi-empirical formulae are given for the section moduli of deck, bottom, and side longitudinals. These maintain existing standards but remove some of the anomalies in the previous tables.

Section 7—Transverse Side Framing. This complex Section is discussed in some detail, with examples of its application, under the headings: Basic Formula for Frames; Main Frames in Bulk Carriers; Panting Frames; Peak Frames; Web Frames in Bulk Carriers; Frames in Engine Room; Frames in Deep Tanks; Frames under Hatch End Beams, etc.; Inertia; Beam Knees and Tank Side Brackets.

Section 8—Deck Beams. The empirical formula given for the section modulus of transverse deck beams takes into account the position of the beam in the ship and the number of deck tiers, as well as beam spacing, span, and loading. There is an alternative formula for use when the beams are subjected to the more concentrated loading due to fork-lift trucks.

Section 9—Double Bottoms. The significant addition to the Rules is in the part dealing with the double-bottom scantlings when heavy or ore cargoes are to be carried with specified holds empty in the loaded condition. A comparatively simple procedure is specified, but investigations still in progress on this complex problem are likely to result in further refinements.

Section 10—Strengthening of Double Bottom Forward. Extensive investigations of damage record cards have shown that the present standards of strength are reasonable. As the risk of damage in this region decreases with increasing draught, the Rules specify that, when the ballast draught forward is 0·03L or less, full Rule increase to the flat of bottom shell plating is required, but that no increase is necessary when the draught is 0·04L or greater. For intermediate draughts the increase is obtained by interpolation.

Section 11—Panting. As compared with the 1966 Rules, the vertical spacing of stringers and tiers of beams has been slightly increased in the peaks. Beam scantlings dependent on ship length and breadth have been given an upper limit, because the loading does not significantly increase above a certain ship size.

Section 12—Stems and Sternframes. There is no essential change in regard to stems; draught is used in preference to length as the basis for the scantlings for sternframes and rudder axles. Large empirical reductions are permitted for sole pieces whose after ends are fixed to rudder posts. Rudder horns are treated as pure cantilevers.

Section 13—Deck Girders and Transverses. The required minimum modulus is determined from a formula involving the span, spacing, and loading of the girder. An additional requirement regarding moment of inertia of girders has been introduced to keep the deck sufficiently rigid to ensure watertightness of hatch covers.

Section 14—Pillars and Non-Watertight Pillar Bulkheads. The formulae allow for residual (e.g. welding) stresses, end constraint, bending moments at pillar ends, and plate buckling.

Section 15—Cantilevers. Rules for cantilevers have been derived by the methods of plastic design. An appendix to the paper shows the derivation of the formulae.

Section 16—Longitudinal Strength in Way of Erections. To calculate the allowable reductions to the section modulus of the main hull girder in way of an erection, a method based on that proposed by Caldwell for single-tier erections (see Abstract No. 12,740, Apr. 1957) has been adopted. The method has been extended to cover multi-tier erections on the basis of shipboard measurements made by Johnson and Ayling (see Abstract No. 18,266, Apr. 1962).

Section 17—Superstructures and Deckhouses. When this Section was prepared superstructure scantlings still had to comply with the requirements of the 1930 Load Line Convention. The Rules of this Section are based upon the following conditions:—

(*a*) That there shall be a graduated change in scantlings between superstructures and small deckhouses, and between deckhouses and exposed casings. (A "deckhouse", unlike a "superstructure", does not extend to the ship's sides.)

(*b*) That scantlings of structures on the freeboard and other decks shall vary inversely with freeboard on a ship of any given length, and directly with length for any given freeboard.

Sections 18 *and* 19—*Bulkheads*. On the assumption that the use of modern notch-tough steels and good detail design prevent the occurrence of brittle and fatigue fractures, plastic design methods have been used to derive the new bulkhead Rules. The Author gives the derivation of the formulae in an appendix.

Sections 20—*Tunnels, and* 21—*Machinery Spaces*. These Sections are basically the same as in the 1966 Rules.

Section 22—*Rudders*. The Author summarises the principles that underlie the new Rules for rudders, which include semi-empirical formulae for rudder couplings and pintles giving results closely in line with the tabulated values in the old Rules. The formula for rudder scantlings also gives results that differ very little from those of the 1966 tables, but some reductions in mainpiece scantlings have been allowed on the assumption that mainpieces will be of fabricated type, well designed and efficiently welded.

Section 23—*Steering Gear*. The requirements of the 1966 Rules have been maintained, except that rod and chain gears have been eliminated. There are also additional requirements for shrinkage allowance on tillers and quadrants, and for the effective sectional area of the key in shear.

Section 26—*Hatchways and Deck Openings*. This Section, which incorporates the relevant requirements of the 1966 International Load Line Convention, deals with directly-secured steel covers for the range of "A" to "B" freeboards, steel pontoon covers for "B" freeboards, and small portable covers in association with beams for increased "B" freeboards only. The Author explains the method used for deriving the requirements for inertia and modulus.

Section 27—*Masts and Rigging*. The Rules for stayed masts have been completely revised; the new method, which is derived directly from first principles, is explained by the Author in an appendix.

Section 28—*Bulwarks, Freeing Ports, Scuppers and Sanitary Discharges, and Side Scuttles*. *Section* 29—*Ventilation, Air Pipes, and Sounding Pipes*. Both these Sections have been rewritten to comply with the requirements of the 1966 Load Line Convention. The basic requirements of Section 28 remain unaltered; Section 29 allows certain slight relaxations from the requirements of the earlier Rules, but steel protection for air and sounding pipes in bulk-cargo spaces is now required.

Section 32—*Welding*. This Section has been considerably revised and rearranged, and requirements for welding higher-tensile steels have been added. Some reductions have been made in leg-length minima for certain continuous fillet welds. Continuous welds are required in after peaks to reduce vibration damage.

Section 34—*Equipment*. The Rules and Tables are the outcome of compromises agreed at a series of meetings held over several years with other Classification Societies.

26,661 **1967 Concept in Tanker Design.** LYNCH, C. M., and WILKINSON, T. T. *American Petroleum Institute, Proceedings of Annual Tanker Conference,* 15-17 May 1967, p. E-1 [11 pp., 7 diag.; and Discussion: 6 pp.]

In their introduction the Authors, of the Sinclair Refining Company, trace the development in hull and machinery design of tankers from just prior to 1930 to the present day. They suggest that continued collaboration (with due regard to proprietary interests) between members of the A.P.I. can greatly assist the development of an optimum tanker for the future. A concept of such a tanker is presented; it is a fully-automated ship capable of long periods of continuous service, and manned by a small, highly trained, technical crew. It is suggested that dry-docking should be at the owner's discretion instead of a regulatory requirement, except in the event of hull or machinery damage. A description and general-arrangement drawings of the proposed ship are given. The principal particulars are:—

Length, o.a.	1,165 ft
b.p.	1,115 ft
Breadth, moulded	160 ft
Depth, moulded	83 ft
Draught, loaded	56 ft
Deadweight, loaded draught	200,000 tons
Cargo capacity (100%)	1,620,000 bulk barrels
Ballast capacity	81,000 tons
draught	30 ft approx.
Propulsion power, normal	40,000 h.p.
maximum	60,000 h.p.
Service speed	17·6 knots

The ship has a bulbous bow and a cruiser stern with horn rudder, a flush weather deck, and main machinery aft. A single six-tier superstructure contains crew accommodation, all domestic spaces (including fresh-water tank), and the navigating bridge (with all-round visibility). The superstructure can be positioned to suit the owner's requirements, but in the drawing it is shown well forward, within 100 ft of the fore perpendicular. A single funnel rises direct from the weather deck over the machinery space, aft of which is an unobstructed landing area for helicopters. The mooring equipment aft, which is hydraulically powered, is below the weather deck, the lines being led through Panama-type fairleads at second-deck level; watertight closures are fitted for the fairleads when at sea. There is a bow-thrust unit just aft of the F.P.

The hull has two continuous longitudinal wing bulkheads between the fore and after peaks; with the side shell they form deep, narrow, ballast tanks and give protection against fire, explosion, and sea-water pollution in the event of collision. An athwartship ballast tank amidships is equipped to act as a roll-stabilising tank when the ship is in ballast. There are four cargo tanks, two forward and two aft of the amidship ballast tank; of these, No. 2 acts as a stabilising tank when the ship is loaded. Each cargo tank is served by a 25,000 bbl/hr self-priming deep-well pump driven by a gas turbine and controlled locally. The tanks are interconnected by bulkhead sluice valves and can be discharged separately, or collectively as a free-flow system. There are no sea connections to the cargo tanks or cargo pumps.

There are two gas-turbine driven combined fire, general-service, and ballast pumps, one forward and one aft, both capable of serving any or all ballast tanks. The forward pump turbine also operates a hydraulic system which powers the anchor windlass and the other forward mooring equipment. There are ten mooring winches in all; like the windlass, they can be remotely controlled from the bridge

The propulsion plant consists of two 20,000-h.p. aircraft-type gas turbines, driving a single controllable-pitch propeller through reduction gearing. The plant is unattended and is remotely controlled from the bridge. The fuel-processing plant is in a separate compartment in the engine room. High fuel costs are at present an undesirable feature of the concept, but this may be alleviated by the use of crude oil from North Africa with its low vanadium and sodium content.

The steering gear is served by hydraulic-power take-offs from the main turbines, and in emergency by the hydraulic system serving the after mooring winches. Two 150-kW gas-turbine driven generators, located in the superstructure, supply electrical power for all domestic services, navigational equipment, and other auxiliaries.

The crew would consist of 12 men, comprising master, three officers, three bridge technicians, one machinery technician, two cadets, one steward and one steward's assistant. Their duties, which are based on a three-watch system, are described. It is suggested that if two crews were available to man the ship in turn, increased efficiency would result compared with the present " leave " system.

Fire-fighting equipment is described; it would be partly automatic, and controllable from the bridge. Four 15-man inflatable liferafts would be carried, but no lifeboats. When in port the main propulsion plant can be shut down and secured.

The project is discussed and criticised by representatives of the U.S. Coast Guard and of Shell International Marine Ltd.

26,662 **Some Thoughts on Mammoth Tanker Design.** ROBERTS, W. J. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15–17 May 1967, p. D-1 [22 pp., 1 ref., 5 graphs, 1 diag.; and Discussion: 15 pp., 4 diag.] See also *Motor Ship*, **48** (1967), p. 263 (Sept.) [4 pp., 5 graphs, 2 diag.]

This paper discusses the various factors which must be considered when designing a very large tanker, and their effects on the design as a whole. It includes a detailed account of the Lloyd's Register design study for a 500,000-ton d.w. tanker of minimum weight and construction cost within the Society's present Rules (see Abstract No. 25,675, Sept. 1967). The considerations underlying the choice of principal particulars for this ship, and its structural design, are explained, with special reference to the use of higher-tensile steel.

The economic advantages of large tankers are discussed. It is shown how, as ship size increases, steel costs rise as a percentage of the total capital cost of the ship; this is an important point, which stresses the need to evolve an optimum steelwork design. Problems connected with the mooring and/or anchoring of these large ships are also discussed in the light of increased size and cost of equipment; unconventional solutions are being considered, e.g. the use of bow and stern lateral-thrust units to replace not only anchors and cables, but also the conventional rudder

and steering gear. The physical problems of surveying ships over 100 ft deep are also considered.

In the Discussion, the views of representatives of the American Bureau of Shipping, Det Norske Veritas, and other interested parties are expressed, and a fairly detailed account (with layout and structural drawings) of Det Norske Veritas' 500,000-ton design study (see Abstract No. 25,676, Sept. 1967) is given.

26,663 **The Fairing of Ships' Lines by Digital Computer** (in German). SÖDING, H. *Hansa*, **104**, No. 16 (1967), p. 1386 (Aug.), and No. 18 (1967), p. 1527 (Sept.) [18 pp., 37 ref., 2 tab., 3 graphs, 9 diag.]

Existing procedures for the mathematical representation of hull forms with the aid of a computer, starting with offsets lifted from 1 : 50 or 1 : 100 drawings, entail certain interpolation and other difficulties. In this article (a thesis accepted by the Technische Hochschule, Hanover, in 1967), the Author, after an historical outline of the development of mathematical representation of hull forms, describes procedures currently proposed and presents a new procedure based on improvements and additions to some of them.

The information is given under the following main headings:—

(*a*) Curve Interpolation.
- (i) Published Methods for the Interpolation of Ships' Lines (use of polynomials).
- (ii) An Advantageous Addition to the Procedure.
- (iii) A System of Equations for Calculating the Deflection Curve of an Elastic Batten.
- (iv) Calculation of the Deflection Curve by a Method of [Natural Elastic] Constants [based on that of Biermann].
- (v) Calculating Ships' Lines by Extending the Method of Constants.
- (vi) Application of the Method of Constants to Interpolation.

(*b*) Representation of and Interpolation on Hull Surfaces.
- (i) Numerical Description of Hull Form (choice of basic offset grid—Lidbro's method is preferred).
- (ii) [Methods of] Smoothing the Surface [and their Drawbacks].
- (iii) An Alternative to Smoothing (the alternative is simulation of the graphical lines construction process).
- (iv) Simulation, by Computer, of the Graphical Design of Hull Lines (a suitable programming language and procedure are described).
- (v) Surface Interpolation.
- (vi) Comparison with Other Methods. (The Author's procedure is applicable to any ship form, including appendages; it is faster and gives better results than other methods, but requires personnel familiar with conventional graphical techniques. Basic offsets are needed only at those points where weights or pegs would be needed in fairing with splines or battens.)

It is mentioned that this study has been extended to the integration of given hydrostatic and other pressure distributions over the mathematically defined hull-surface. This extension leads to calculating methods for the

forces and moments due to fluid pressure and for the determination of these quantities after alterations in trim.

26,664 **The Caledon Timber Carrier.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 404 (22 Mar.) [2 pp., 1 tab., 6 diag.]

To meet a demand for ships that will carry not only bulk cargoes, but also timber, logs, and other cargoes that call for stowage in large unobstructed rectangular holds, the Caledon Shipbuilding and Engineering Co. has produced two suitable single-deck designs of 16,000–17,000 tons d.w. These have a vertical forefoot, a transom stern, and a clearwater stern frame with spade rudder. The conventional hopper-sided self-trimming hold construction of a bulk carrier has been replaced by "open-ship" construction, with large twin (side-by-side) hatches at each of the four holds, a deep centre-line member, and narrow side tanks for water ballast. There is also a pair of large ballast tanks between No. 1 hold and the collision bulkhead. The accommodation is as far aft as possible, to increase the length of weather deck available for cargo stowage; this should eliminate the practice of stowing timber deck cargoes at uneven heights to control the c.g. position and trim. Moreover, the design is such that the bale-to-grain ratio is greatly increased. The holds have natural ventilation and (like the engine room) a CO_2 fire-extinguishing system. Accommodation for a complement of 47 is heated and ventilated mechanically.

The two designs (Proposals 1 and 2) differ mainly in the cargo-handling arrangements adopted. Proposal 1 has four 15-ton swinging derricks, and Proposal 2 has four 15-ton fixed deck cranes. A separate design incorporating heavy-lift gear is also available. For propulsion machinery, the shipbuilders offer a geared Pielstick 16PC2V of 8,000 b.h.p. at 520/135 r.p.m.; a Sulzer 6RD68 of 8,000 b.h.p. at 150 r.p.m.; or a B. & W. 6K62EF of 8,815 b.h.p. at 140 r.p.m. (These are all maximum continuous ratings.) The direct-drive installations involve a longer engine room (by three frames), and a consequent reduction in cargo capacity.

The principal particulars of the two designs are as follows:—

	Proposal 1	*Proposal 2*
Length, o.a.	148·29 m (486 ft)	152·86 m (501·5 ft)
b.p.	137·16 m (450 ft)	141·73 m (465 ft)
Beam, moulded	22·40 m (73·5 ft)	22·40 m (73·5 ft)
Depth, moulded	12·34 m (40·5 ft)	12·34 m (40·5 ft)
Summer draught	8·79 m (28·8 ft)	8·86 m (29·1 ft)
Deadweight, corresponding	16,050 tons	17,000 tons
Timber draught	9·04 m (29·7 ft)	9·09 m (29·9 ft)
Deadweight, corresponding	16,750 tons	17,750 tons
Capacities		
grain	21,250 cu m (750,000 cu ft)	22,080 cu m (780,000 cu ft)
bale	21,000 cu m (740,000 cu ft)	21,800 cu m (770,000 cu ft)
Water-ballast capacity	5,050 cu m (5,100 tons)	5,450 cu m (5,500 tons)
Speed, smooth-water trials at 90% m.c.r.	14·75 knots at 29·7 ft draught	14·5 knots at 29·9 ft draught
Range	13,000–14,000 miles	

26,665 ***Port Chalmers*, World's Largest Refrigerated Cargo Vessel.** *Shipp. World & Shipb.*, **161** (1968), p. 943 (June) [10 pp., 1 tab., 12 diag., 7 phot.], and *Shipbuild. Shipp. Rec.*, **111** (1968), p. 853 (21 June) [6 pp., 1 tab., 5 diag., 6 phot.]

The twin-screw refrigerated motor-ship *Port Chalmers*, recently delivered to the Port Line by the Linthouse (Alexander Stephen) division of Upper Clyde Shipbuilders, is by far the largest (and is one of the fastest) of ships of this type. A sister ship, the *Port Caroline*, is now completing at the same yard. These two liners will connect the U.K. and north European ports with Australia and New Zealand. Principal particulars of the *Port Chalmers* are:—

Length, o.a.	612·2 ft
b.p.	570 ft
Breadth, moulded	81 ft
Depth, moulded, to upper deck	48·5 ft
Draught, loaded	35·5 ft
Deadweight	19,710 tons
Gross register	16,283 tons
Displacement	29,860 tons
Lightweight	10,150 tons
Block coefficient	0·628 at loaded draught, 0·598 at 29 ft draught
Cargo capacity, refrigerated	606,940 cu ft
liquid	460 tons
general (bale, including liquid cargo)	236,000 cu ft
containers (20 × 8 × 8 ft)	71
Propulsive power, max. continuous rating	2 × 13,000 s.h.p.
service	2 × 12,000 s.h.p.
Service speed	21·5 knots

The ship has a raked stem, bulbous bow, and cruiser stern. The engine room and superstructure are three-quarters aft. To enable high speed to be maintained in bad weather, the forecastle is long and, from the region between Nos 1 and 2 holds, the upper and forecastle decks have considerable sheer, the bows being heavily flared before they merge into a return curve to avoid excessive width at deck level. There is a long poop. The ship is designed especially for palletised and other unitised loads (containers can be carried on deck), and the cargo spaces have been given as regular a shape as possible; obstructions which could impede fork-lift trucks have been minimised. Apart from the forward sheer mentioned, and the camber of the weather deck, all decks are without sheer or camber.

There are five holds forward of the engine room and two aft. Refrigerated cargoes can be carried in Nos 2 to 6 holds and their lower tweendecks, in No. 2 upper tweendeck, and in lockers (port and starboard) in Nos 3 to 6 upper tweendecks. General cargo is carried in Nos 1 and 7 holds and in the centre spaces of Nos 3 to 6 upper tweendecks. There are three edible-oil tanks in No. 1 hold. " Dangerous " cargoes can be carried.

The holds were insulated by Gregsons (of London and Tyneside); they feature flush vertical and overhead surfaces incorporating ventilation-ducts

and light-fittings, and a new type of corrugated mild-steel dunnage which is fitted over the on-deck insulation. This dunnage is harder wearing than wooden sheathing (over £10,000 per annum may be saved in consequence), takes up less space, allows air circulation along the corrugations, and can support fork-lift trucks with an all-up weight of 12 tons; it is readily removable.

Apart from six Trans-Roto hatch covers on the weather deck, the covers are of the electrically-operated Screw-Torq type; these 11 Screw-Torq hatch-covers are of high-tensile steel. All covers (apart from those on No. 7 tweendeck) are fully insulated. Each of the Screw-Torq covers consists of six sections arranged as three pairs; one pair stows at one end of the hatch and the other two at the other end. Apart from some Grace Line installations, this is the first ship to have Screw-Torq covers; some information is given on their operating mechanism. (See also Abstract No. 25,027, Jan. 1967, for information on both Trans-Roto and Screw-Torq covers.) All 17 covers can be opened and closed from the weather deck by one man. For this purpose, a portable 24-V control box is plugged into a socket in each of the hatch coamings in turn (there are three of these control boxes, enabling the covers of three holds to be operated simultaneously). The control box has a switch for selecting Open, Neutral, or Close, and another (with seven positions) for selecting which cover, and which sections of that cover, are to be operated. There is also individual push-button control for each weather-deck cover. As a safeguard, electrical interlocks prevent a lower cover from being opened before, or closed after, an upper one. Extra-quick Cleveland securing-dogs assist the simplicity of these operations The covers were supplied by Cargo Dynamics (G.B.) Ltd.

The cargo-refrigeration plant, supplied by J. & E. Hall, has five 170-h.p. eight-cylinder R.22 Veebloc compressors. Spirally-wound gill-type coolers, Axia high-efficiency axial-flow units, and, for the upper-tweendeck lockers, Axia extended-spindle fans, are used in the installation. The refrigerated spaces have automatic temperature-control; Dayton control valves and Negretti & Zambra pneumatic temperature-controllers are largely used. The spaces are monitored by a Richardsons Westgarth 280-point data-logger which can operate in three modes: Manual, Auto, and (for full print-out) Log.

Five-ton traversing cranes are provided between Nos 1 and 2, between Nos 2 and 3, and between Nos 4 and 5 hatches; two 5-ton fixed cranes are mounted between Nos 6 and 7 hatches. They are all of the Stothert & Pitt electro-hydraulic " Stevedore " type. There are also four 10-ton and two 15-ton derricks; these are operated by Siemens (U.K.) electric cargo-winches, and have separate electric winches for topping and slewing; when used as swinging derricks, they can be topped and slewed when under load. A 25-ton Thomson swinging derrick is mounted between Nos 3 and 4 hatches. All derricks and cranes have a large outreach. Other deck equipment mentioned includes twin electric windlasses, two self-contained electric capstans, and two self-tensioning electric mooring-winches, all supplied by Siemens (U.K.). Brown Bros electro-hydraulic rotary steering-gear is fitted. A Decca steering-control system, in conjunction with an Arkas automatic pilot, is used; the operator can use either hand control (when standing) or foot control (when seated, his hands thus being free for engine manœuvres).

The two main engines are Clark-Sulzer 6RD90 Diesels. Their bridge-control system is a development of the shipbuilder's Mahout system. Woodward synchro-phasing gear is provided (see Abstract No. 24,646, Sept. 1966). Two John Thompson exhaust-gas boilers provide a total of 16,750 lb/hr of steam; there are two Cochran Spheroid boilers for harbour use. At sea, electrical load is normally carried by a 1,100-kW Brotherhood Siemens turbo-alternator supplied as a packaged unit. To meet extra-heavy electrical loads at sea, and for harbour use, there are five 410-kW Siemens alternators driven by Rolls-Royce Diesels. Load-sharing between the turbo-alternator and the Diesel sets is automatically controlled. Further information is given on the main and auxiliary machinery installations. The engine room was designed with the aid of a 1/10-scale model; this technique is used by the builders for all new construction. The central console is on a flat forward of the engines.

The wheelhouse and chartroom are combined. The steering console includes the Mahout main-engine controls. An engine-room telegraph, intended for emergency use only and operated by push-button, is provided. Life-saving equipment includes two 98-person Viking Marine g.r.p. motor lifeboats and two 25-man inflatable rafts.

There is accommodation for a crew of 55, plus some extra accommodation for personnel under training. Officers have single cabins arranged to allow their wives to accompany them. The other crew members, except cadets and boys, also have single cabins. There is accommodation for 12 passengers.

26,666 ***Sheaf Crest*** **A 14,830-ton Liberty Replacement Ship from Doxford.** *Motor Ship*, **49** (1968), p. 183 (July) [4 pp., 1 tab., 13 phot.], and *Shipbuild. Shipp. Rec.*, **111** (1968), p. 865 (21 June) [3 pp., 2 tab., 1 graph, 2 diag., 1 phot.]

The *Sheaf Crest*, built at the Sir James Laing yard by the Doxford & Sunderland Group for the Sheaf Steam Shipping Co. (Managers: W. A. Souter & Co.), was delivered in June 1968, less than 15 months after being ordered. She is the first Doxford Liberty-replacement ship (see Abstract No. 25,890, Nov. 1967), but several non-standard features are incorporated to meet the owner's specification. These include heavier cargo-handling gear (10-ton derricks and 5-ton winches) for handling timber cargoes, an additional alternator to cope with this heavier gear, an extra boiler and an extra purifier, the provision of an air-conditioning system (the ship will be used for world-wide service), and an engine-room console with centralised control and instrumentation (including a mimic diagram for the alarms).

The principal particulars of the *Sheaf Crest* are:

Length, o.a.	462·3 ft
b.p.	445 ft
Breadth, moulded	71 ft
Depth, to upper deck	38·25 ft
to second deck	28·25 ft
Displacement, loaded	18,675 tons
Draught, loaded	28·1 ft
on tonnage marks	25·4 ft

Deadweight	14,830 tons on loaded draught; 12,740 tons on tonnage-mark draught
Register tonnage	9,392 (gross), 6,739 (net), on loaded draught; 6,389 (gross), 3,992 (net), on tonnage-mark draught
Grain capacity	776,310 cu ft
Bale capacity	722,270 cu ft
Service speed	14½ knots

The tonnage-mark draught, 25·4 ft, is the same as that of the standard version (wrongly given as 28·4 ft in Abstract No 25,890). The loaded draught is about 4 inches less than that of other Liberty replacements of similar tonnage on order, and 16 inches less than that of some Liberty-replacement designs.

The second-deck hatch covers can be used, in conjunction with light portable "modular" bulkheads, as longitudinal grain-feeders, thus eliminating the usual wooden divisions. The five tweendeck hatches can be prepared for loading grain in just over three hours, as compared with about 24 hr when conventional wooden feeders are used. These hatch covers were specially designed by the builders in collaboration with W. A. Souter & Co., and are of the pontoon type, with four sections to each of the larger openings; they are of steel box-construction, and are stiffened to allow the use of fork-lift trucks. The covers open athwartships, and can be fixed vertically (for grain cargoes) in alternative positions. The *Motor Ship* article includes photographs of these arrangements. The upper-deck hatch covers are MacGregor single-pull units.

The main engine is a Doxford 58J4 Diesel (see Abstract No. 26,630, Aug. 1968), with a maximum continuous rating of 6,000 b.h.p. at 140 r.p.m. Both articles give a description of the engine and the machinery arrangements; that in the *Motor Ship* includes photographs of the engine and its components, and that in *Shipbuild. Shipp. Rec.* includes performance curves and a diagram showing the dimensional requirements for overhauling. The latter article also gives a loading scale and information on the ship's outfit, and mentions that equipment for further Doxford Liberty-replacement ships will, in general, be selected from the same outfit range.

26,667 ***Finlandia*** **- Finnish-Built Passenger and Car Liner—Some Design Considerations.** *Shipp. World & Shipb.*, **160** (1967), p. 1500 (Sept.) [14 pp., 2 tab., 18 graphs, 17 diag., 9 phot.]

The vehicle/passenger ferry *Finlandia* was generally described in Abstract No. 25,893 (Nov. 1967). The present article gives further details of the ship's construction and passenger accommodation and, in particular, describes the methods adopted in the design stage to determine an optimum hull and machinery layout.

Using a preliminary arrangement drawing in conjunction with an advanced computer program (which had been developed for the purpose), weights, capacities, hull coefficients, stability, propulsion and resistance values, and building costs were computed. To fulfil certain requirements, some parameters were automatically varied or iteratively computed. Results of these preliminary parametric studies are discussed and shown

graphically, and the main factors which determined the final design are stated.

The positions of VCG and LCG for a calculated empty-ship weight and for the empty ship in the inclining condition are shown in a table. Intact stability was compared by the computer with different accepted criteria, which are listed and illustrated graphically. Watertight subdivision is arranged on a two-compartment basis to enable deck passengers to be carried if required.

Three model hull forms were tested in the Trondheim tank to investigate the influence on speed of decreasing waterplane coefficient coupled with increasing beam, but with the damage-stability margin remaining constant. The model-test results are shown graphically; the individual characteristics of the models are compared in a table with those of the lines finally adopted. Model tests were also made with double and single rudders, and with different arrangements of the two Pleuger bow-thrust units. A bulbous bow was tested, but was not adopted as its length would have been restricted by the bow-ramp arrangement.

The results of full-scale spiral, circle, and zig-zag steering trials, and also speed trials, are discussed and illustrated; the speed trials showed an increase of 0·5 knot over the figure predicted by the model experiments.

Model tests were carried out with the Muirhead-Brown "controlled passive" tank stabilising system. This is not typical of passive tanks in general, as it is tuned to a shorter period to suit the controlled system. The results obtained are discussed and shown in roll response curves.

Two 600-b.h.p. Pleuger bow-thrust units with fixed blades were preferred to one 1,200-b.h.p. unit, as the smaller units allow better immersion and are more readily available from manufacturers; a braking device prevents propeller rotation in a seaway when the units are not being driven.

The numerous advantages which led to the choice of a multi-engine geared medium-speed propulsion plant are discussed, as are its few disadvantages. The main particulars of the four Sulzer Z40/48 propulsion engines (described in Abstract No. 25,922, Nov. 1967) are given in a table; in the *Finlandia* their output has been somewhat derated from the design values. Test-bed performance results of one of these engines are shown graphically. Reasons for choosing the Lohmann & Stolterfoht transmission system, which incorporates Pneumaflex couplings, are given. The reduction gears and couplings can withstand continuous stresses about 50% in excess of the normal, to permit operation in hard-ice conditions, and are fitted with automatic overload-protection devices which can be preset. Pitch control of the KaMeWa propellers can be effected electrically in the event of failure of the pneumatic system, and pitch is automatically reduced should the main engines become overloaded.

Details of the extensive measures taken to reduce noise levels throughout the ship, both from installed machinery and from propeller-excited vibration, are described. Noise and vibration measurements were recorded on trials and are shown graphically. Particular attention was paid to the machinery control room, and this is illustrated by a graph showing the results of noise tests made in the control room before and after insulation. During the trials some oscillation occurred between two of the main engines when coupled as a pair to the reduction gear, but this was overcome by altering the governor excitation frequency by changing to stiffer buffer springs.

26,668 ***Star III*—Design and Construction.** TOHER, R. A., and DAWSON, J. H. *A.S.M.E., Paper No. 66 WA/UNT 8, presented* 27 Nov. 1 Dec. 1966 [5 pp., 1 diag., 3 phot.]

Star III, the third in a series of small submarines developed by the Electric Boat Division of General Dynamics for undersea testing and research, has been designed for an operating depth of 2,000 ft and a maximum speed of 6 knots. The machined spherical (5½ ft diameter) pressure cabin, of HY-100 steel, has five windows. Although this two-man craft displaces only 20,000 lb, it has a payload (available for scientific purposes) of 1,000 lb. The craft is equipped with a manipulator arm, lights, precision navigation equipment, television, and sonar. Its design and construction are described in some detail; they represent a departure from the normal techniques used at the Electric Boat Division.

26,669 **Self-Dumping Log Ship.** *Maritime Reporter*, **29** (1967), p. 28 (15 July) [½ p., 2 diag.]

A design has been patented by Lunde Carriers, San Francisco, for a "self-dumping log ship" in which sea water provides additional buoyancy for the cargo. Nippon Kokan Kaisha (NKK) have been granted sole rights for the construction and sale of the craft in Japan. The unconventional design is expected to reduce both building and cargo-handling costs.

The craft, known as the Lunde Logger, resembles a swim-ended dumb barge; it has a full-width flat transom and only one transverse bulkhead (to the forepeak). It has narrow side tanks (which provide buoyancy in the unloaded condition), and a single bottom which has openings to the sea, allowing the hold to be kept flooded. Logs are loaded into the flooded hold until the designed load waterline is reached; in this condition the submerged and floating logs provide adequate buoyancy not only for their own weight, but also for part of the weight of the logs above the waterline.

Discharging is effected by lowering a panel in the after swim and allowing the logs to float out; the panel is the full width of the hold and is hinged at its lower edge just above the turn of the swim.

Chains, which are secured to the bottom of the hold and to athwartship beams at deck level, enable the logs to be segregated for parcel shipment, and are also used for securing the cargo in transit and for controlling the dumping process.

A profile drawing illustrates the dumping process, and there is a sketch of the loaded craft with tug standing by.

26,670 **Ferro-Cement Construction for Fishing Vessels.** FYSON, J. F. *Fishing News International*, 7 (1968), p. 51 (Apr.), p. 39 (May), and p. 30 (June) [9 pp., 9 ref., 11 phot.]

In these three articles, the Author discusses the economics, design, and construction of fishing craft built of a material known as "ferro-cement" or "ferro-concrete"; this was first introduced by the Italian structural engineer P. L. Nervi in the early 1940s and developed in Great Britain under the name of Seacrete by Windboats Ltd. of Wroxham (see Abstract No. 26,671, this issue).

The material consists of a fine sand/cement mixture reinforced by a number of layers of soft iron-wire mesh. It was first used by Nervi to produce thin slabs capable of withstanding high loads and with a flexibility

not previously obtainable without severe cracking of the cement mortar. Thicker slabs (up to 10 cm, i.e. 4 in) with similar properties were obtained by using one or more central layers of ¼-in or ⅜-in iron rods, to which the outer layers of mesh were tied. The material must be finished off to a thickness only slightly greater than that of the reinforcement. With a weight of reinforcement of 25–32 lb per cu ft, the ultimate tensile strength of the material is 5,340 lb/sq in, which is five times that of the unreinforced mixture and comparable with that of wood. Moreover, the material is homogeneous and therefore, unlike wood, of equal strength in all directions.

The outstanding advantage of ferro-cement as a boat-building material is the low cost of the materials required to produce a strong hull. The constituents are sand, cement, certain concrete additives, wire mesh, ¼-in steel rod, tie wire, and ordinary (water or steam) pipe material. Labour forms a relatively high proportion of the total cost, but much of it need not be highly skilled. Moreover, the capital investment required to produce ferro-cement hulls in a boatyard is small. Apart from equipment for mixing, applying, and curing the cement, a small welding machine, a pipe-bending machine, and drills, together with wood-working machinery and tools for deckhouses and interior joinery, are needed. An advantage over g.r.p. is that no moulds are required and that one-off hulls can be economically produced.

The material is lighter than steel, and in craft above about 40 ft in length, the hulls are no heavier than those of wood or g.r.p. The surface should be painted, for appearance and as an additional protection for the mesh reinforcement close to the surface of the mortar. Maintenance is limited to touching up the paint. Structural damage due to collision or grounding is usually less than with a wooden hull, and is much easier to repair.

In the second article the Author describes in some detail the method of construction of a ferro-cement hull. The design is first lofted in the usual way. The line of stem, keel, sternpost, and horn timber is then made up from ¾ to 1½ in o.d. water pipe (depending on the size of the boat), and the frame outlines are formed by bending similar piping to the requisite shape. The deck camber is formed of similar pipe, which is welded to the frame pipe. The skeleton framework must be well supported in such a way that it does not distort under the heavy loads imposed on it in subsequent stages of construction.

When all the framing is in place and either welded or tied to the backbone structure, longitudinal and vertical rods of ¼-in iron or high-tensile steel are bent round the whole hull at 2 to 4 in centres, following the hull lines, and tied into place with soft iron wire. Six to eight layers of mesh are then tied to the framework with iron wire, half of the layers being inside and half outside the framework. Ordinary galvanised or plain ½-in bird netting is suitable for the mesh. The cement mixture is then applied, various additives (especially pozzolan) are used to control quality and reduce water content. The Author describes the various stages in this process; plastering techniques are used for most of the hull, but the keel section is poured. The last stage is curing of the cement, which, when controlled by water sprays, may take 21 days. By the use of a steam generator, the curing period can be reduced to about 24 hours.

Various modifications of the building process have been developed,

notably by Ferro-cement Ltd, of New Zealand, and the Fiber Steel Corporation of America; the latter firm has evolved a mould method for producing stock hulls to a standard design.

In the third article, the Author discusses the design of ferro-cement hulls. For boats 35 to 55 ft long, the hull thickness should be not less than ¾ in; for lengths up to 80 ft this should be increased to 1 in. The weight of reinforcement should be not less than 24 to 36 lb per cu ft, which is equivalent to about 2 lb per sq ft of surface area in ¾-in thick material. The total weight of the material is then 9 or 10 lb per sq ft. It is thus heavier than wood for a sawn-frame oak fishing boat of length 40 to 45 ft, but lighter for larger hulls. To obtain the strongest hull for a given shell thickness, the form should be fully curved, with no flat runs. Structural bulkheads and floors are formed integrally with the hull; internal ferro-cement gussets can be used for attachment of other partitions, etc.

The properties of the material and the particular techniques of its application to boat construction, many of which are explained by the Author, enable strong, economical hulls to be produced which are well suited to the needs of the fishing industry. See also the following two Abstracts.

26,671 **Seacrete Method Developed by Progressive British Yard.** BURGESS, J. *Fishing News International*, 7 (1968), p. 44 (May) [2 pp., 1 diag., 1 phot.]

This is a general account of Seacrete as a boatbuilding material (see also Abstract No. 25,607, Aug. 1967). It includes general-arrangement drawings for two 47-ft fishing boats recently built by Windboats Ltd for operation from Aden (see also following Abstract). Seacrete is particularly suitable for teredo-infested tropical waters.

26,672 **Seacrete [Fishing] Boats for Somalia [and Aden].** *Fishing News International*, 7 (1968), p. 55 (Apr.) [½ p., 1 phot.]

INDUSTRIAL AND ECONOMIC INFORMATION

26,673 **Tanker Chartering Practice—Past, Present, and Future.** GLANDER, O. G. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15–17 May 1967, p. F-1 [8 pp.; and Discussion: 2 pp.]

The Author reviews the history of tanker rate reference tables, and describes their modernisation and simplification. A new "World Scale" table covering 50,000 rates has been evolved by the New York and London offices of the Association of Ship Brokers and Agents and the International Tanker Nominal Freight Scale Association. It was to become effective on 1 Jan. 1968.

BOILERS AND STEAM DISTRIBUTION

26,674 **Operating Experience with Single Boiler Ocean Tank Vessels.** MCTAGGART, T. J. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15–17 May 1967, p. I-1 [12 pp., 1 tab., 1 diag., 2 phot.]

This paper discusses the first two years' operating experience with the

single main boilers of the 66,700-ton d.w. tankers *Esso Houston* and *Esso New Orleans*, both owned by the Humble Oil and Refining Company, Houston, Texas. These ships are powered by cross-compound geared steam turbines developing 19,000 s.h.p. at 104 r.p.m. (See also Abstract No. 25,347, May 1967.)

Before the ships were built, a close survey was made of the performance and reliability of D-type and sectional-header marine boilers which had bee. in service from four to 21 years. The results of this survey are discussed; down-time data are given in a table. The boiler finally chosen was of a type which had proved particularly reliable in the past, and, by agreement with the manufacturer (the Combustion Engineering Co.), incorporated certain features which were thought desirable. It is designated Type V2M-VS, and its features are listed as follows.

(1) " Bent Tube " type rated at 150,000 lb/hr at 600 lb/sq in and a total superheat of 860 F. (2) Full automatic firing control over entire range. (3) Furnace heat release not exceeding 70,000 BTU per cu ft of furnace volume. (4) Internal desuperheater of 130,000 lb/hr. (5) External steam air preheater only. (6) Economiser with cast-iron extended heating surfaces; tubes welded into headers and hand-hole plates seal-welded. (7) Round hand-hole plates except for one oval plate at each end of headers. (8) Tube spacing $2\frac{1}{4}$ in clear. (9) Large external downcomers to give circulation ratio of 16 to 1. (10) Electrically-operated safety valve. (11) Steam pilot-operated superheater relief valve. (12) Retractable steam soot blowers. (13) 60-in diameter steam drum. (14) 1,600-lb drum level-gauge glasses. (15) Furnace floor tubed, and refractory rated at 3,000 F. Tangent waterwalls – refractory kept at about 500 F by screen of tangent tubes. (16) Wide-range rotary-cup burners on starboard side of boiler (reasons for choosing this type are given). (17) Electrical control system for combustion and burners. (18) Two secondary air blowers with stepless automatic control. (19) Ports in target wall for observation of burners. (20) Opening to boiler furnace for engine-room waste disposal. (21) Comprehensive fail-safe equipment. (22) Two turbine-driven multi-stage centrifugal feed pumps (instead of the usual three).

There is a 750-kW turbo-generator in the engine room, and a 350-kW Diesel generator in a compartment outside. The Diesel set is for use during boiler shut-downs and for powering essential services should the turbo-generator fail; in this latter event it starts automatically and enables the ship to maintain a speed of about 14 knots. To meet SOLAS requirements, there is also a 25-kW Diesel set arranged for automatic start. Some other features of the simplified propulsion system are discussed, and the percentage savings in piping, valves, etc. for main steam supply, boiler feed and condensate, and combustion control are listed.

During their first two years' service the *Esso Houston* and the *Esso New Orleans* have had no boiler breakdowns at sea, and only minor repairs have been necessary in port. Some problems, connected chiefly with the rotary-cup burners, did arise, but these are being overcome in ways which are described. Bunker C fuel, mostly from Venezuela, has been used throughout in both ships. Boiler inspections after one and two years in service revealed no serious defects.

The Author draws attention to the importance of correct chemical

treatment of boiler feed water, and recommends that the regular testing of samples by a qualified analyst be strictly observed.

There are photographs of the V2M-VS boiler and the W. N. Best Combustion Equipment Co.'s rotary-cup burner.

26,675 **Combustion of Crude Oil in Ships' Boilers.** NORRIS, R. O., and BASSETT, R. S. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15-17 May 1967, p. H-1 [44 pp., 9 ref., 20 tab., 4 diag., 3 phot.; and Discussion: 4 pp.]

This is the full text of a paper which formed the basis of the article covered by Abstract No. 26,005 (Dec. 1967).

DIESEL AND OTHER I.C. ENGINES

(*See also Abstracts No.* 26,666, 26,667, *and* 26,678)

26,676 **Development of Compact 1,000-h.p. per Cylinder Engine [the Fairbanks Morse 38A20 Range].** ANTONSEN, A. K., and DAHLUND, E. L. *A.S.M.E., Paper No. 66-DGEP-20, presented* 24-28 Apr 1966 [10 pp., 1 tab., 1 graph, 27 diag., 13 phot.]

The 38A20 range of compact medium-speed (400 r.p.m.) opposed-piston engines includes in-line and V models, and covers the power range 6,000 to 18,000 h.p. Dual-fuel versions burning either oil or gas, and reversible marine versions burning residual fuels, are available. The present paper gives a detailed description, together with some account of prototype performance testing. Various problems encountered in the development of these engines, and the ways in which they were overcome, are explained. A water-flow model was used in studying air and gas flow patterns.

See also Abstracts No. 23,351 (Aug. 1965) and 24,810 (Nov. 1966), and item 12 in Abstract No. 25,177 (Mar. 1967).

26,677 **Tests to Examine High-Pressure Pulse Charging on a Two-Cycle Diesel Engine.** WOODS, W. A. *A.S.M.E., Paper No. 66-DGEP-4, presented* 24-28 Apr. 1966 [15 pp., 7 ref., 5 tab., 33 graphs, 8 diag.]

This paper describes a comprehensive series of tests carried out at the University of Liverpool on the B.S.R.A. experimental opposed-piston two-stroke Diesel engine; this is a 1/5 scale model of a marine engine of this kind, and runs at about five times the full-scale speed. The objects of the tests were to investigate the effects produced by wide changes in engine load or i.m.e.p. (0 to 160 lb/sq in), supercharge pressure (3 to 15 lb/sq in gauge), and exhaust-piston lead (0 to 10°) on: (*a*) engine performance and (*b*) available energy content of the exhaust gas. The exhaust turbine was simulated by means of a convergent nozzle, and the exhaust-gas energy was calculated from pressure and temperature diagrams recorded in the exhaust pipe near the nozzle. The transient temperatures were measured by fine-wire resistance thermometers (see Abstract No. 21,825, Aug. 1964).

The main conclusions from the test results are as follows. The engine performance and the exhaust-gas energy available at the nozzle are both almost independent of the exhaust-piston lead over the range covered. The engine power depended directly upon the fuel flow up to the smoke limit, but this limit could be extended by using higher boost pressures. The exhaust-gas energy increased with increase in engine load or in

boost pressure. The theoretical gain in exhaust-gas energy associated with the change to large exhaust-piston leads was lost owing to throttling at the exhaust ports. This loss is thought to occur in the exhaust belt, which had been designed for a non-supercharged engine. The tests demonstrated that the possible gains in blowdown energy are high; further design and experimental work in this direction should be carried out. An approximate method for calculating the exhaust-gas energy from a pressure diagram and a thermocouple reading has been developed and is found to give accurate results. A simple theoretical cycle analysis is given in the appendix.

The work described was sponsored by B.S.R.A.

MARINE POWER INSTALLATIONS (GENERAL)

26,678 **A Comparison of High-Powered Single-Engine and Multi-Engine Plants for the Propulsion of Merchant Ships.** ZINNER, K. *A.S.M.E., Paper No. 67 DGP 2, presented* 23 27 Apr. 1967 [12 pp., 12 ref., 3 tab., 1 graph, 8 diag., 4 phot.]

This paper is mainly concerned with a quantitative comparison between possible propulsion installations for a single-screw bulk carrier requiring about 20,000 s.h.p. The engines concerned are of various M.A.N. designs; some particulars of their design features, and of service experience with them, are given. The installations are:

(i) A direct-coupled K9Z 86/160 E engine (20,700 s.h.p. at 118 r.p.m.), running on 1,500-sec. fuel.

(ii) Two 18-cylinder VV 40/54 engines (9,780 h.p. each at 400 r.p.m.), with clutch couplings and gearing, running on 400-sec. fuel. For an account of this four-stroke trunk-piston engine design, see Abstract No. 26,426 (May 1968).

(iii) Ten VV 22/30 ATL 16-cylinder four-stroke engines (2,200 h.p. each at 900 r.p.m.) with D.C. electric transmission; they run on Marine Diesel fuel. The newer VV 23/23 TL engine (2,500 h.p. at 1,500 r.p.m. in a 12-cylinder version) would have given a cheaper and more compact installation, but insufficient service data were available.

(iv) Twenty VV 16/18 TL 16-cylinder four-stroke engines (1,060 h.p. each at 1,500 r.p.m.) with D.C. electric transmission; they run on gas oil.

(v) A common frame carrying 120 six-cylinder automotive-type engines (D2146 MT; 180 h.p. each) together with their D.C. generators; the fuel is gas oil. This scheme was not considered practical, but was included in the comparison because it has sometimes been suggested that ships should be propelled by numerous small engines.

Detailed comparisons are made in respect of initial cost, weight, space, floor area, height of engine room, fuel cost, lubricant cost, and maintenance and spare-part costs. The results are summarised in a schematic diagram and discussed. The main conclusions are as follows. Diesel-electric installations are only justified in special circumstances. It is not economic, even from the standpoint of maintenance cost, to split installed power among a large number of small engines. High-speed engines are,

in general, uneconomic because they cannot burn heavy fuel. Geared medium-speed engines running on heavy fuel have become an attractive alternative to the large low-speed engine on grounds of savings in initial cost, weight, and space, and greater versatility in operation. The operating and maintenance costs are, however, somewhat higher. It is for the shipowner to decide the relative importance of these factors for any particular ship project.

26,679 **A New Concept of Electric Ship Propulsion.** HARVEY, L. M., and FULMER, R. D. *Marine Technology*, 5 (1968), p. 158 (Apr.) [6 pp., 1 tab., 1 graph, 6 diag., 1 phot.]

This is a paper presented to the Soc. N.A.M.E. (New England Section) in May 1967.

A homopolar generator (also known as a unicyclic or acyclic generator) is a rotating machine designed so that the relation between the effective moving conductor and the stationary magnetic field does not vary during a revolution. In principle, pure D.C. (no ripple) can be tapped off through slip-rings and brushes or other contact collectors; if connected to an external D.C. supply, the machine functions as a motor. Apart from the simple Faraday disc, such a machine can take the form of a solid cylindrical rotor of conducting material, surrounded by a stator drum incorporating annular field conductors and two annular current collectors; the same pattern of field and current-path lines will then appear in any axial plane. The outstanding advantage over conventional D.C. machines is the elimination of the commutator and the associated size limitation.

The development of homopolar machines was inhibited by the fact that, even when run at high speeds, they remain essentially low-voltage high-current devices; it was difficult to devise satisfactory low-resistance collectors for these conditions. However, in the mid 1950s U.S. General Electric introduced the eutectic alloy of sodium and potassium (known as NaK) as the current-transfer medium. This liquid metal has electrical conductivity 100 times greater than that of graphite brushes, density about 85% and viscosity about 50% of the values for water, and good heat-transfer properties; it readily wets other metals. Large generating plants using this technique have been operating successfully for several years; some particulars are given. Two 12 MW continuous-duty generators (72 V, 165,000 A) have been built and tested, and a commercial range is available. In homopolar motors, contact must be maintained at zero speed; this can be achieved by completely flooding the stator rotor gap with NaK retained by shaft seals.

The Authors set out the relevant theoretical considerations, and proceed to compare the particulars of 15,000-h.p. turbo-electric propulsion schemes using (*a*) commutator D.C. machines and (*b*) homopolar machines. In both cases the propeller is driven at 175 r.p.m. by a direct-coupled motor; the commutator motor has a double armature. Scheme (*a*) uses two generators driven through gearing at 600 r.p.m.; (*b*) has a single acyclic generator directly driven at 5,000 r.p.m. The weights and dimensions of the machines are very much smaller for (*b*) than for (*a*); motor efficiency is the same for both (94%), but the acyclic generator is more efficient than the commutator machines (98% instead of 93%). The ratings for each machine (or motor armature) are 1,000 V, 12,000 A in (*a*), and 76 V, 157,000 A in (*b*). Scheme (*a*) has very little overload

capability, whereas the acyclic machines are hardly susceptible to damage by overload or short-circuit. It is therefore proposed that the heavy currents of (b) should be carried by solid copper or aluminium buswork (the machines being located as close to each other as possible) with no switchgear whatever; control would be through the excitation of the two machines, and a suitable system giving full speed control in both directions is described. Buswork power losses can be kept below 0·67% without incurring a serious weight penalty. The acyclic machines are sealed, and work in an atmosphere of pure nitrogen; this is primarily to avoid oxidation of the NaK, but incidentally excludes salt air and other contaminants. Maintenance operations are briefly discussed.

It is pointed out that a D.C. transmission permits the use of a unidirectional constant-speed prime mover, which can also drive a ship's service alternator.

AIR CONDITIONING, VENTILATION, AND REFRIGERATION

(*See Abstract No.* 26,694)

AUXILIARY EQUIPMENT AND MACHINERY

(*See also Abstract No.* 26,695)

26,680 **Central Oil-Hydraulic Stations in Large Ships** (in German). TORNOW, H., and RÜCKGAUER, N. *Hansa*, **104**, No. 18 (1967), p. 1545 (Sept.) [7 pp., 1 graph, 7 diag., 9 phot.]

The Authors first describe and discuss current practice in the design of hydraulic systems and their components, with special reference to centralised "hydraulic stations" in ships. This general information is followed by a detailed description of the hydraulic installation in the 19,000-ton (gross) LPG/chemical carrier *Paul Endacott* (for a description of this ship, see Abstracts No. 21,254, Apr. 1964, and 22,302, Nov. 1964).

This ship has a 1,250-h.p. hydraulic station located amidships. The station is equipped with nine electrically-driven hydraulic pumps of the variable-displacement axial-piston type; they serve a large number of hydraulic motors, through an extensive network of piping. This arrangement allows easy access to the various components; this is particularly important where they are in spaces subject to explosion hazard. Other equipment in the station includes the feed pumps (for supplying the hydraulic pumps), together with the servo-control pumps, oil reservoirs, filters, and coolers.

Three of the nine pumps supply the power for driving the three compressors (including one standby) provided for cargo reliquefaction, each compressor being driven by a 110-h.p. hydraulic motor. Each of these three pumps is regulated through a hydraulic servo device, and its hydraulic circuit includes a relief valve, a feed-pressure valve, and a cut-off pressure-switch. This system can be controlled manually from the bridge, compressor-room, or engine-room, or automatically; the circuit operation is described.

Five of the nine pumps in the hydraulic station serve the cargo-pump hydraulic motors, of which there are ten of 70 h.p. each and four of 12 h.p. each. Each hydraulic pump forms part of a unit which includes various regulating valves. There are five main centrifugal cargo pumps on the port side of the ship and five to starboard; if desired these two groups can

be supplied with oil independently of each other, though normally all the motors are fed from a hydraulic ring-main. The motors can be put on or off line from the bridge through a pneumatic arrangement, and are normally served by four of the hydraulic pumps; the fifth hydraulic pump can serve the port or starboard group if required, and also serves as a standby for the fan group used for circulating cold nitrogen in the spaces around the cargo tanks.

The nitrogen-circulation system can keep the cargo at a temperature of 7 C (19 F); it is used when the cargo is of a type unsuitable for reliquefaction, and is manually controlled. The fans are driven by ten 10-h.p. hydraulic motors, arranged in a ring circuit and served by the remaining pump of the nine previously mentioned. This pump is similar to the hydraulic pumps of the cargo-pumping system.

The ship's four mooring winches and two combined windlasses mooring-winches are hydraulically driven, the motors being supplied from the cargo-pumping ring-main circuit.

The article contains much additional information on this hydraulic installation and its components and pneumatic control arrangements. Circuit diagrams, and photographs of the equipment, are included.

26,681 Current Status of Saline Water Distillation. PORTER, J. W. *A.S.M.E., Paper No. 67 UNT 5, presented* 30 Apr. 3 May 1967 [9 pp., 12 ref., 2 tab., 3 graphs, 6 diag.]

Three methods of saline water distillation multistage flash, multiple-effect, and vapour compression—are in use today. Other methods, including hybrids of the first two, are under study. Dual-purpose plants, which produce both power and water, appear to be the most economic. Several technological problems require further study; in particular, formation of scale in the piping, determination of the best materials for condenser tubes, and improvement of heat-transfer rates. This paper explains the principles of the methods mentioned, and reviews the present situation.

26,682 Electrodialysis Past, Present, Future. KIRKHAM, T. A. *A.S.M.E., Paper No. 67 UNT 8, presented* 30 Apr. 3 May 1967 [9 pp., 9 ref., 2 tab., 2 graphs, 6 diag., 1 phot.]

The electrodialytic principle of water purification makes use of special membranes which, when exposed to an electric potential gradient, allow only anions (or only cations) to pass. The nature of the process makes it very suitable for conversion of brackish waters. More than 100 installations are in operation in various parts of the world. Future improvements may well lead to economic conversion of sea water by electrodialysis. The present paper is a fairly detailed review of the subject.

AUTOMATION, INSTRUMENTS, AND CONTROL DEVICES

26,683 U.K. Manufacturers' Approach to the Specification and Design of Marine Automation Systems. VALENTINE, F. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia,* 22 24 Nov. 1967 [11 pp., 1 ref.]

The Author first draws attention to the co-operation between the Marine

Automation Committee of the British Electrical and Allied Manufacturers Association (BEAMA), the U.K. Chamber of Shipping, B.S.R.A., and the leading Classification Societies. Such co-operation is necessary in order to produce economical and reliable automation systems meeting ship operators' requirements. It is noted that BEAMA is preparing a revised edition of its "Guide to the Specification and Purchase of Electronic Equipment for Industrial Systems"; this will cover marine applications, giving advice on environmental and safety considerations, planning, installation, testing, and maintenance.

The desirability of automation specialists being brought in at an early stage of ship and engine-room design, and of a single contractor being given overall responsibility for the automation systems, is pointed out. Only thus can full integration be achieved. The equipment should be modular, to permit rapid repair by replacement of sub-assemblies rather than components (faulty components take time to trace). The use of a standard range of equipment and systems is very desirable. Other aspects discussed are the selection and training of seagoing personnel (automation manufacturers' courses will become essential, although clear and comprehensive manuals must be provided for use aboard each ship), and reliability (the "mean time between failures" is not a sound criterion for marine automation equipment, since the consequences of a component failure may be slight and repair by replacement very rapid; a better one is time percentage availability).

26,684 **Thyristors in Motor Control. A Survey of Current Practice.** *Electrical Review*, **183** (1968), p. 87 (19 July).

The articles in this survey are as follows.

Why Choose a Thyristor Drive? STRETTON-DOWNES, J. N. [2 pp., 2 phot.]

Which Scheme to Select? BOWLER, P. [3 pp., 4 diag., 3 graphs]

The Author discusses: D.C. motor drives from A.C. supplies, (*a*) up to 5 h.p., and (*b*) from 5 to 100 h.p.; D.C. motor control with D.C. supplies; A.C. motor drives; the cycloconvertor; a quasi-square-wave self-excited invertor; pulse-controlled convertors; slip-ring induction-motor control; new machines (using thyristor circuits to replace or supplement mechanical commutators—in the latter case the commutator acts as a "dead" switch).

Far-Reaching Improvements to Multi-Machine Drives. SCHOFIELD, J. R. G. [1 p., 1 phot.]

The Author discusses the use of static thyristor convertor units in place of conventional Ward-Leonard or Kramer sets.

Switching with Semiconductors. FELTBOWER, B. [1 p., 1 diag.]

Power thyristors are now available for voltages up to 2,000 V (peak) and current ratings of several hundred amperes. Overload and surge protection, modular construction of static contactors, reliability, and cost (which, for contactors, is still considerably higher than that of electromagnetic units) are briefly discussed.

Developments and Applications. [3 pp., 1 diag., 5 phot.]

This covers: an extruder drive for submarine-cable manufacture; the

Cressall Thyristat for control of A.C. fan drives; the Photain combined socket-outlet motor-control unit for small appliances; the Bercotrol tension-control system for rollers in textile plant; Bruce Peebles' "Vress" thyristor-convertor D.C. drives (25 to 300 h.p.) with comprehensive surge protection and accurate speed control; roller table drives using cyclo-convertors ("S" series of Laurence, Scott & Electromotors); the Siemotron variable-speed drive for fractional-h.p. A.C. motors (this incorporates transistors and Hall-effect generators); Mather & Platt's wide-range (over 100 : 1) variable-speed drive for D.C. motors of 5 to 3,000 h.p.

26,685 **Bridge Control and Power Metering, with Special Reference to Turbine Ships.** STENOW, A. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia*, 22–24 Nov. 1967 [21 pp., 3 graphs, 8 diag., 6 phot.]

A short account is first given of an advanced bridge-control system for steam-turbine installations which has been developed jointly by three Swedish firms (ASEA, Kockums, and Stal-Laval). It is of solid-state electronic type, and incorporates testing equipment (including a plant simulator). This is followed by a more detailed description of the ASEA "Torductor R" magneto-elastic torque transducer (torquemeter), which is suitable for permanent installation on propulsion shafting. (See also Abstract No. 24,353, June 1966.) The aspects covered include: principle and mechanical design; characteristics; calibration; measuring accuracy (better than ±1% under reasonable service conditions—accuracy is not impaired by dirt, moisture, or vibration); electrical equipment. Descriptions are also given of digital equipment in which the Torductor output is combined with signals from other transducers (tachometer, flowmeter) to obtain readings of power and specific fuel consumption.

DECK MACHINERY AND CARGO HANDLING

(See also Abstracts No. 26,665, 26,666, *and* 26,680)

26,686 **A Study of the Factors in Cargo Gear Selection.** HAMMOND, N. *Ship and Boat*, **21** (1968), p. 18 (May) [2 pp., 1 tab., 1 graph]

Except for vessels carrying cargoes that are loaded and discharged by shore-based equipment, the turn-round time of a ship in port depends mainly on the nature and capacity of its cargo-handling gear.

There is a tendency for union-purchase derricks to be superseded by the ship-borne crane. In 1962 about 20% of ships entering service had at least one deck crane, and about 70% of the ships so equipped were bulk carriers. At about the same time the Velle and Hallén swinging derricks were introduced, being used mainly for general and bulk cargoes respectively. For handling containers the Carron Unloader has been developed. However, a study of recently-built tramp vessels designed to operate between the U.K. and Mediterranean ports has shown a strong preference for derricks. Less than 2% of the vessels considered had deck cranes only, about 5% had both derricks and cranes, and about 90% had derricks only. The remainder either had no cargo gear or were fitted with special equipment for particular trades.

The Author gives a graph showing the total capital costs of derricks,

and of fixed and travelling deck cranes, over the capacity range up to 20 tons per unit. For a ship of about 3,000 tons gross with eight 3/5-ton derricks, the total capital cost of the cargo gear would be £48,000 to £50,000. An alternative installation of four fixed slewing deck cranes would cost 45 to 50% more. The Author considers, however, that the cranes could handle significantly more cargo per hour than the derricks. This conclusion is based on cycle-time observations in European ports; average times for each phase of the crane and derrick cycles are tabulated. Little information is available on which to base a comparison of maintenance costs for derrick and for crane installations, but it is probable that they do not differ greatly; these costs vary considerably with the ports at which repairs or surveys are carried out.

26,687 **Prospects of Development of Remotely-Controlled Hatch Covers.** KAJNČ, F. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia,* 22–24 Nov. 1967 [13 pp., 1 ref., 1 tab., 18 diag.]

The development of mechanised hatch covers, and of locking and sealing systems for them, is first reviewed. Short accounts are given of the hydraulically-operated "Lift-Lock" system, and of the Itomatic inflatable seal (for which see Abstract No. 23,607, Oct. 1965). Control of hydraulic hatch covers from remote positions has hitherto not been favoured owing to the additional piping and power losses involved. The difficulty can be overcome by using electrical control circuits acting on solenoid valves in the hydraulic loop; the extent and complexity of the hydraulic system can then be greatly reduced, and it can be completely protected from the weather. The correct sequencing of operations can be provided for on the electrical rather than the hydraulic side. It is noted that the French MacGregor company has installed such a control system, using 24 V solenoid valves, in the passenger ship *Pasteur*, which has 12 "Folding" and "Multifolding" covers. The system has been in use for over a year and has proved very satisfactory. The Author envisages extension of the scheme to stern and side doors in pallet carriers, etc., and to the large, heavy, and closely-spaced hatch covers of container ships.

26,688 **Automatic Mooring Winches and Remote Control of Anchor Windlasses.** ŠVERER, Z. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia,* 22–24 Nov. 1967 [19 pp., 2 graphs, 12 diag.]

The mechanisms of self-tensioning mooring winches with steam and with electric drive are explained in some detail, special attention being paid to load-balance arrangements which limit line-tension peaks to not more than 20% over the nominal rating. The use of synthetic rope on mooring winches in the automatic mode is inadvisable (rope life being reduced by adhesion of adjacent turns), unless the main load is taken on a warping head and the rope coiled on a special reel, which may be installed below deck and whose drive is electrically synchronised with that of the winch itself.

Finally, a hydraulic system for remote push-button control of a windlass brake (e.g. from the bridge) is described.

26,689 **Proposal for a Container Supporting Arm.** *Holland Shipbuild.*, **16** (1967), p. 63 (Oct.) [1 p., 1 diag.]

The majority of standard-size containers are only strong enough to be

stowed in up to six tiers, i.e. to an overall height of about 48 ft, and hold depths are generally based on this fact. However, to make the best use of a large container ship, whose optimum dimensions may well merit a hold depth greatly in excess of 48 ft, seven or more tiers would have to be carried. In this case some form of intermediate support must be fitted to relieve the weight on the lower tiers. A method of doing this without recourse to tweendeck hatches is described.

The proposed supporting equipment, illustrated by a diagram, consists of straight hollow steel arms of rectangular section (the dimensions proposed for 20-ton containers are 8 in deep and 4 in wide, with a material thickness of 0·375 in). There is one such arm for each corner of the cell: it is pivoted at one end (outside the container stowage space) through a heavy horizontal gusset plate secured to the flanges of the guide angles of adjacent cells at a pre-determined height. The gusset plate is extended transversely between and beyond these adjacent guide angles to form a support for the other ends of the arms when they are swung out from their retracted positions into diagonal cross-corner positions, ready to receive the corner castings of containers. The arms may be rotated manually or, preferably, in groups of four by remote control from the deck. Scantlings of the guide angles would be suitably increased to withstand the directly superimposed loads, and additional cross-bracing fitted to prevent buckling.

The savings in weight and cost by adopting this supporting-arm method have been estimated for two ships designed for carrying 500 and 800 30-ft containers respectively. The advantages may be summarised as follows: (1) A reduction in cost of £50,000 to £70,000 per ship by the elimination of tweendeck hatches and covers (any loss in longitudinal strength can be compensated for elsewhere). (2) A complete cell can be loaded or unloaded without interruption. (3) Cell guide angles can be permanently fitted in one length. (4) The overall depth of the ship can be reduced and its stability improved (when carrying a given number of containers), because the depth of the arms is much less than that of a hatch cover. (5) A saving of 250 to 350 tons in light weight of a ship of given size and speed. (6) Lower maintenance costs and power requirements for the operation of supporting arms as compared with power-operated hatch covers. (7) No space required for stowage of hatch covers, thus allowing maximum clear opening of the weather deck. (8) Supporting arms can be designed to sustain any desired size and number of containers.

26,690 **S.S. *Mobil Acme*—Specialty Products Carrier Utilising Hydraulic Power for Cargo Pumping.** SUTHERLAND, F. X. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15–17 May 1967, p. 1-47 [13 pp., 7 diag., 3 phot.]

The paper describes a hydraulically-powered cargo-pumping system in the 19,000-ton d.w. *Mobil Acme*. This steam-turbine tanker was built in 1960 by Alexander Stephen as a clean-products carrier with a conventional cargo-tank layout. She was converted to a specialised tanker for lubricants, solvents, and chemicals at Sasebo, Japan, in 1963. The conversion involved the fitting of two additional transverse bulkheads, which subdivide the original Nos 2 and 3 tank sets into smaller compartments with capacities ranging from 2,000 to 4,000 bulk barrels. Nine of these compartments have been provided with individual deepwell pumps.

For the remaining tanks, the original piping and pumping plan was re-arranged to form six entirely segregated tank-groups, each with its own pump (four in the after and two in the forward pump room). Thus 15 different liquids can be carried in lots ranging from 2,000 to 35,000 bulk barrels. The tanks have been given epoxy or zinc coatings to suit the special products normally carried in them.

The reasons for choosing hydraulic drive for the nine new Byron Jackson deepwell pumps are stated. The low-pressure (500 lb/sq in) hydraulic system, which was developed by the Rockwell Manufacturing Co., is described with the help of a circuit diagram. The system temperature is maintained by one heat exchanger with a 500 gal/min cooling capacity. Fluid temperature is kept to a maximum of 110 F. A 300-gal capacity combined reservoir and pressurising unit, handling 5 gal/min, maintains suction pressure to the main hydraulic supply pumps; the pressure is pre-set high enough to prevent pump cavitation and air-locking.

The hydraulic power plant is in the forecastle and consists of two 250-h.p. electrically-driven rotary pumps (shown in a photograph) of the double-acting constant positive-displacement type. Each deepwell pump is driven by a vertically-mounted hydraulic motor operating on the same principle as the hydraulic pumps, and developing 70 h.p. at 1,750 r.p.m.; the speed is controllable from zero to maximum. Deepwell cargo pumping is controlled from a central station amidships, where a panel displays cargo-discharge pressure gauges, hydraulic discharge and suction gauges, individual motor tachometers, and remote speed-control equipment for the pump motors.

This hydraulic equipment has operated satisfactorily in the *Mobil Acme*, and the system may be extended to incorporate three more of the ship's cargo pumps.

The paper also discusses the hydraulic pumping equipment installed in the 15,780-ton d.w. *Mobiltest I*, which, at the time of writing, was moored off Nigeria and used as an unmanned tank storage vessel. Her conversion for this service was carried out in Montreal in 1966, and called for a system which could be quickly activated by a work-gang air-lifted at short notice from the mainland. The medium-pressure (up to 1,600 lb/sq in) hydraulic system is made up of Racine Vickers-Armstrong equipment and is Diesel powered; it drives the anchor windlass, six mooring winches, three bilge pumps, ventilation blowers, and two deepwell cargo pumps. The hydraulic circuit is described and shown in a diagram.

The Author concludes by discussing the numerous advantages of hydraulically-powered deepwell cargo pumps, and refers in particular to the possible saving in initial cost by the elimination of pump rooms, the ability of hydraulic motors to function under water, their inherent safety when operating in an explosive atmosphere, and the relative freedom of hydraulic systems (as compared with steam-turbine or Diesel drives) from corrosion problems. A possible future development is location of both pump and drive motor at the bottom of the cargo tank instead of on deck.

26,691 **Experience with a Direct Transfer [Oil] Cargo System.** MAYBOURN, R. *American Petroleum Institute, Proceedings of Annual Tanker Conference*, 15/17 May 1967, p. 1-13 [14 pp., 2 graphs, 3 diag., 1 phot.]

The paper describes a direct-transfer cargo system which has been

installed in five 64,000-ton and two 111,000-ton d.w. B.P. tankers. It was extensively tested in the *British Commerce*, the first of the smaller ships, which was completed in 1965. The cargo-handling specification for this ship called for three steam-turbine driven cargo pumps, each with a salt-water capacity of 2,000 tons/hr; a conventional piping system would have required lines of about 16 in diameter and valves of a size very near the limit for manual operation.

The direct system is based on transferring cargo from tank to tank through specially-designed hydraulically-operated rectangular penstock-type valves located at the bottoms of the transverse and longitudinal bulkheads; the valves, which are described and shown in a photograph, have a vertical sliding movement. This arrangement enables all cargo-pump suctions to be sited in the aftermost centre tank, and long runs of large-diameter lines are avoided. One 12 in × 18 in valve between each wing tank and its centre tank, and three 21 in × 24 in valves in each transverse bulkhead separating adjacent centre tanks, are necessary to ensure an unrestricted flow at maximum loading and discharging rates while maintaining a reasonable trim. The pumping layout is described with the help of diagrams. As the pumps are self-priming, separate stripping pumps and lines are not required. However, there is a 12-in ring main for handling ballast, tank cleaning, topping up the cargo, and for discharging a tank should a valve fail to open; this ring main can also be used to discharge a particular tank or tanks if more than one grade of cargo is being carried.

Loading and discharging procedure and methods of maintaining trim are described in detail. The three 16-in diameter direct loading lines are each rated at 3,000 tons/hr, and the 12-in ring main at 1,500 tons/hr through each leg. The main steam-turbine cargo pumps are each rated at 1,550 tons/hr (for oil of s.g. 0·85), and can deliver crude at 110 lb/sq in at the ship's rail. The pumps normally run at 1,750 r.p.m. and have a degree of automatic control; their ancillary equipment (which includes a gas and vapour separation system on the suction side) is described with the help of a diagram. The cargo pumps in the 111,000-ton d.w. ships (e.g. *British Admiral*; see Abstract No. 23,892, Jan. 1966) are electrically driven, and have a more comprehensive remote valve-control system.

There is no ballast pump in the *British Commerce*, but the permanent ballast tanks are fitted with eductors driven by the tank-cleaning pump in the engine room; this pump has a dual rating of 400 tons/hr for tank cleaning and 1,000 tons/hr for ballast duties. In practice much of the ballast is run in or run out. Two eductors, each of 250 tons/hr capacity, are fitted in the pump room for tank cleaning; they each require 190 tons/hr of driving fluid at 130 lb/sq in, which is obtained from one of the main cargo pumps. All hydraulically-powered valves in the cargo tanks and pump room are operated by integral linear actuators working at 2,500 lb/sq in. The hydraulic-power system, and the actuator design, are described.

Some of the more important advantages of the direct-transfer system over a more conventional arrangement are listed. They include: (1) Saving in weight of about 200 tons. (2) Reduction in first cost of about £15,000. (3) Improved performance due to elimination of suction-line losses. (4) Simplified operating technique. (5) Reduction in

maintenance and repair costs. (6) Improved oil flow through the centre tanks. A disadvantage could be that the lack of large pipelines tends to reduce efficiency when carrying several grades of cargo, but this is not serious in a large crude-oil carrier.

The Author concludes by mentioning some of the problems which had to be overcome when fitting the bulkhead valves and actuating systems in the earlier ships. These arose chiefly from lack of experience in the shipyard with the comprehensive hydraulic systems, non-availability of long lengths of hydraulic piping (resulting in the use of too many joints), and the lack of a sufficiently high standard of cleanliness during the work. It is suggested that, in future systems, the direct loading line should be placed one tank space further forward to enable maximum loading rates to be maintained for a longer period. It is also suggested that, in order to indicate when a tank is empty, the deck-mounted mechanical float-type ullage gauges should be fitted at the after ends of the tanks instead of at their centres. This has already been done in two ships with good results; trim corrections must be applied when topping up, until the ship is on an even keel.

26,692 **Underwater Oil Terminal.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 868 (21 June) [1 p., 1 diag.]

A brief description is given of an underwater crude-oil storage tank, with a capacity of about 87,000 tons, which is to be built for the Dubai Petroleum Co., a subsidiary of the Continental Oil Co. The tank is to be anchored, with its base on the sea bed, at an offshore site in the Persian Gulf, 65 miles off Dubai; this will eliminate the need for shore installations and pipelines thereto, and large tankers unable to move into the shallow waters will be able to "secure near the storage tank" to load crude oil, even in severe conditions.

The tank (shown in a sketch) will consist of the storage portion, shaped like a dome and resting on the sea bed, and a cylindrical column protruding from the top of the dome and extending above water. The total height is 205 ft and the water depth 155 ft; the base of the tank is 270 ft in diameter, and the tank weight is 12,500 tons.

The tank design was developed by the Chicago Bridge & Iron Co., who are to build the tank in a specially-designed dry dock at Dubai. The tank will be floated out through a series of locks and thence to its station 65 miles offshore, sunk to the sea bed, and anchored. The same company installed a much smaller offshore storage tank (of different design) in the Gulf of Mexico a few years ago.

The tank will be operated on the oil water displacement principle, i.e. as oil is produced it will be pumped into the tank and will thus force water out, and, as a tanker is being loaded, water will flow into the tank to replace the oil being pumped out.

The development of the design, over a period of four years, involved the solution of problems concerning: the seaworthiness of the tank during towing; the stability of the tank during its self-submergence; the dry dock for building the tank; sea-water corrosion over a period of at least 20 years; winds, waves, and currents at the site; contamination of the oil by salt "pick-up"; contents gauging; oil gas separation; and removal of wax and sludge. In some of the development work, 1 : 50 scale models were used.

VIBRATION AND SOUND-PROOFING

(See also Abstract No. 26,667)

26,693 **Wave-Excited Hull Vibration—Measurements on a 47,000-ton d.w. Tanker.** BELL, A. O., and TAYLOR, K. V. *Shipp. World & Shipb.*, **161** (1968), p. 412 (Feb.) [7½ pp., 6 ref., 2 tab., 15 graphs, 4 diag.]

The work described was carried out by B.S.R.A.

During an earlier investigation on an all-aft 47,000-ton d.w. tanker, it was observed that the two-node mode of vertical vibration was frequently excited by some form of wave action (see Abstract No. 24,300, June 1966). In certain sea conditions the maximum vibration stresses occurring in the deck amidships were greater than those due to vertical wave bending moments, especially in short steep waves. In addition, the vibration was translated into continuous fore-and-aft motion in the navigating bridge structure, which was at times unpleasant for the ship's personnel and caused some trouble with the navigating instruments.

Special sea trials were carried out with the object of throwing some light on the factors influencing this vibration. For reasons of improved propulsive performance the ship was later fitted with a ram bow, and a series of repeat measurements was then made to ascertain whether this had altered the vibration characteristics. The present article covers both sets of trials. The ship was in normal ballast condition. Both vibratory and wave bending stresses were measured by a 100-in base strain gauge on the upper deck (centreline, amidships). Resistance strain gauges were located at the centres of large panels in the bow structure (both before and after alteration), to provide more information on the wave excitation.

The measurements showed that the amplitude of the vibration stresses increased approximately as the square of the ship's speed; it was a maximum in head seas and negligible in following seas. The observations appear to be consistent with the vibration being excited by impulsive forces acting on the bow.

The highest vibration-stress range experienced was 3 tons/sq in, which was recorded during the repeat trials on the ship when fitted with the ram bow. This compares with a maximum of 1·5 tons/sq in for the original bow form. However, it cannot be concluded that the fitting of the ram bow increased the level of vibration, because of differences in the sea states encountered.

26,694 **Simplify Your Calculations for Quiet Fan Systems.** TRICKLER, C. J. *Air Condit. Heat. Vent.*, **64** (1967), p. 69 (Jan.) [8 pp., 1 ref., 4 tab., 8 graphs, 1 diag.]

The sound ratings of fans are published by the manufacturers, but do not tell the prospective user whether the installation will be satisfactorily quiet.

Starting from first principles, the Author presents a quick method for calculating the approximate sound level for the fan system as a whole. A work sheet for a typical problem is set out in detail; this problem is the determination, for a dual-duct and mixing-box system, of whether a silencer is required and, if so, how much attenuation is needed and where the silencer should be placed.

As the method does not take into account the effects of duct lining,

vaned elbows, or unusual duct design, it is not recommended for systems having such features. Ships' fans are not specifically mentioned.

26,695 **The Use of Gases Other Than Air in the Acoustical Testing of Model Compressors.** WELLS, R. J., and MCGREW, J. M. *A.S.M.E., Paper No. 67 GT 27, presented* 5 9 Mar. 1967 [14 pp., 10 ref., 2 tab., 13 graphs, 12 diag.]

An account is given of theoretical and experimental studies of the feasibility of simulating the acoustical and aerodynamic characteristics of a full-scale axial-flow air compressor by a scale model working with another gas in a closed-loop system. If the speed of sound is lower in this gas than in air, the tip speed for a given Mach number is correspondingly lower. Freon 12 or 23 was used in the main experiments. Correlation was attained between model and prototype for both acoustical and aerodynamic performance. Agreement was obtained with a previously published generalised acoustic prediction curve. A tentative acoustic scaling law is developed which permits rapid prediction of full-scale sound power levels from measurements of the sound power levels produced by a model.

CORROSION, FOULING, AND PREVENTION

26,696 **Marine Coating is Tested on Tug Boat.** *Welding J.*, **46** (1967), p. 852 (Oct.) [1 p., 2 phot.]

A new coating system under test consists of three to four mils of aluminium flame-sprayed on to the steel surface, followed by three organic top coats. The first top coat is a polyvinyl butyral wash primer to seal the metallised surface; the second is a thin coat of clear vinyl, to prevent leaching of the chromate in the primer; and the final coat is of tributyl tin oxide (TBTO) to prevent fouling. The system (with, however, two coats of TBTO) is being tested on a tug operating in New York harbour and the Hudson River, where the salt water and winter ice result in the average tug needing dry docking for coating maintenance about every 12 to 18 months. The surfaces of the tug were prepared by sand-blasting; the aluminium was applied with wire-metallising guns. Similar coatings (but having only 3 mils of aluminium and no TBTO) are in "perfect condition" after 12 years' testing, by the American Welding Society, in the sea off N. Carolina. The test with the tug is the first commercial test of the system, and is being conducted by a leading oil company (unnamed).

The advantages claimed for the new system include excellent resistance to scraping and abrasion, and ease of maintenance; fouling is easily removed (it does not become firmly attached, even without a TBTO coating). The aluminium gives a form of cathodic protection in which the anode is distributed over the surface; if small areas of steel become exposed through mechanical damage, there is still a high anode cathode ratio and the exposed steel is protected with negligible sacrifice.

26,697 **The Behaviour of Aluminium- and Zinc-Coated Sheet Steel, Compared with that of Aluminium and Zinc Sheet, when Exposed to Industrial, Rural, and Marine Atmospheres** (in German). FRIEHE, W., MEUTHEN, B., and SCHWENK, W. *Stahl u. Eisen*, **88** (1968), p. 477 (16 May) [7 pp., 13 ref., 4 tab., 12 graphs, 10 phot.]

An account is given of outdoor exposure tests on sheet-steel specimens

variously protected by 40-micron (0·0016-in) sprayed and hot-dipped aluminium and zinc coatings, and by 40-micron rolled aluminium cladding. Specimens of aluminium and zinc sheet were also tested, for comparison. The specimens were exposed to industrial, rural, and marine atmospheres for up to ten years; they were inspected and weighed periodically.

The results are reported in detail and discussed, with conclusions and recommendations. Of the steel-based specimens, the best anti-corrosion performance was given by those with the hot-dipped aluminium coatings and those with the aluminium cladding; the Al loss rate was too small to measure in the rural and marine atmospheres, and amounted to about 5 microns over ten years in the industrial atmosphere (where, however, slight rusting occurred at local penetrations). Zinc coatings were removed at about 1 micron/year in rural and marine atmospheres, and about 4 microns/year in the industrial atmosphere; additional protection by painting is recommended.

26,698 **The Effects of Microstructural Variations on Stress-Corrosion Cracking Susceptibility of Ti-8Al-1Mo-1V Alloy on Marinised Gas Turbines.** DANESI, W. P., SPRAGUE, R. A., and DONACHIE, M. J. *A.S.M.E., Paper No. 67 GT 5, presented* 5–9 Mar. 1967 [9 pp., 10 ref., 5 tab., 4 graphs, 11 phot.]

Sea salt can cause titanium alloys to crack under tensile stress. If long-term high-temperature operation of titanium-alloy gas-turbine components in salt atmospheres is desired, means must be found to eliminate or control this tendency. The Authors describe in detail their stress-corrosion testing of a titanium alloy; the results are tabulated. The effects of various heat treatments, and the consequent alterations in microstructure, on the degree of resistance to stress corrosion are illustrated and discussed.

OPERATION AND MAINTENANCE

26,699 **Shipboard Systems Costs: A Functional Analysis of Work Aboard Ship. Summary Report.** *Publication of the Shipboard Mechanisation and Manpower Committee of the Maritime Transportation Research Board, National Academy of Sciences—National Research Council, Washington, D.C.* (Nov. 1967) [50 pp., 16 ref., 15 tab., 1 diag.]

Previous reports and other papers have dealt with information obtained from observations on board various ships taking part in this functional analysis of shipboard work, in which 20 ships were involved; see, for example, Abstract No. 26,296 (Mar. 1968). The present Report summarises the project as a whole.

MISCELLANEOUS

26,700 **Permanent Magnets.** *Engr*, **224** (1967), p. 448 (6 Oct.) [1 p., 3 tab.]

This article consists of summaries of three papers presented at a conference on magnetic materials and their applications, held by the Institution of Electrical Engineers in London, 26–28 Sept. 1967. These papers deal, respectively, with the following topics.

Flexible Magnets. EDWARDS, A.

James Neill & Co. (Magnets and Steel) Ltd, of Sheffield, produce flexible permanent magnets in the form of strip or sheet consisting of a

rubber or thermoplastics material heavily loaded with ferrite particles. Such material can be cut to any desired size or shape. Injection mouldings can also be produced if the quantity required justifies the cost of tooling.

Typical properties are tabulated for several weight percentages of ferrite. If desired, the attractive force can be greatly increased by adding mild-steel pole pieces. The material is dark brown, but, in the case of sheet, a layer of polished p.v.c. of any colour can be bonded to the surface; this layer will accept screen-printing or riting. It is suggested that small pieces of the material should be used as symbols or indicators on charting or recording boards with a steel surface (see also Abstract No. 26,558, July 1968). Many other applications of flexible magnets are mentioned in the paper itself.

Flexible Anisotropic Magnets. BOWMAN, G. G., SKELTON, R. G., and WALSH, L.

The Authors (of the Plessey Co. Ltd) explain how the magnetic properties in a preferred direction can be improved by suitably orienting the ferrite crystals. This can be done by mechanical methods (rolling or extrusion), by magnetic action, or by a combination of these techniques. Typical properties of such flexible magnets are tabulated.

Permanent Magnets for Future Transport. POLGREEN, G. R.

Simply-shaped (brick or ring form) ferrite magnets of large size and very high coercivity can now be produced. As the ferrite ceramic is a good electrical insulator, massive blocks of it can be magnetised by an intense but short D.C. pulse, at negligible energy cost. The permanence of the magnetisation is much superior to that of metal magnets. A further advantage of the ferrite magnet is its relatively light weight. The material is brittle and difficult to work; it can be protected by steel pole pieces, which also overcome the limitation to low flux densities. The properties of typical ferrite and metal magnets are contrasted in a table.

Experimental work has shown the technical feasibility of using stationary or rotating ferrite magnets for the fields of electrical machines, especially those of homopolar (Faraday-disc) type; advantageous application to the drives of electric road vehicles is envisaged. Another scheme is magnetic levitation of guided vehicles above a "track" of ferrite bricks.

26,701 **[Sea-Bottom] Slope Stability Factors to Consider in Offshore Drilling Operations.** MONNEY, N. T. *A.S.M.E., Paper No. 67-UNT-3, presented* 30 Apr.-3 May 1967 [12 pp., 45 ref., 4 tab., 3 graphs, 7 diag.]

26,702 **Aircraft Electrical De-Icing [by the Dunlop Foil-Element System].** *Aircr. Engng*, **40** (July 1968).

The individual articles in this feature are as follows:

An Introduction to Aircraft Ice Protection Systems. HARPER, T. W. p. 6 [1 p., 1 phot.]

Aircraft Electrical Ice Protection Systems. FOSTER, R. A. p. 7 [1½ pp., 4 diag.]

Materials and Manufacturing Techniques. NEWMAN, M. A. p. 8 [2 pp., 5 phot.]

Heating Element Design Features. SMITH, A. R. J. p. 11 [1½ pp., 7 diag.]

Control Systems for Aircraft Electrical Ice Protection. GAULD, J. F. A. p. 12 [1½ pp., 2 diag., 1 phot.]

Harrison & Sons, Ltd., by Appointment to Her Majesty The Queen, Printers, 20-22 Bedfordbury, London, W.C.2